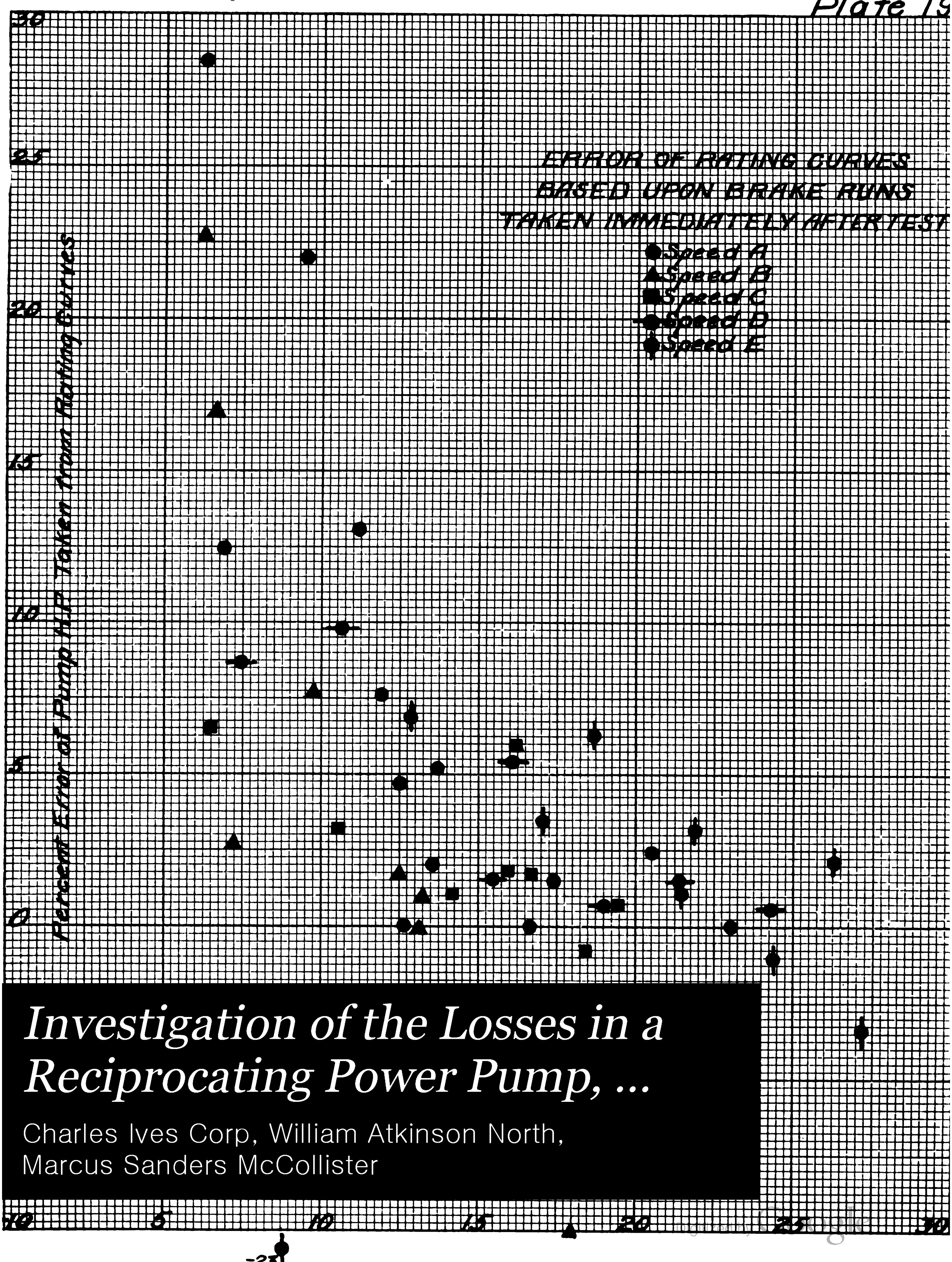

Investigation of the Losses in a Reciprocating Power Pump, ...

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INVESTIGATION OF THE LOSSES IN A RECIPROCATING POWER PUMP INCLUDING A SPECIAL STUDY OF THE VALVE ACTION

A Thesis Submitted

by

Charles Ives Corp

For the Degree of Master of Science

Mechanical Engineering Course

and

William Atkinson North

and

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For the Degree of Civil Engineer

University of Wisconsin

1911

CONTENTS

LIST OF DRAWINGS AND CURVES

INTRODUCTION

The reciprocating or plunger type of pump is used to a greater extent than any other form for the handling of water and various other fluids. It is found from the smaller, hand operated, sizes to the immense pumps used in supplying water for our large cities.

In the smallest sizes the question of the efficiency or the losses in the pump itself is an unimportant one because the total energy supplied is insignificant. But the saving effected by a few percent change in the efficiency, as the size increases, makes the analysis of the losses an important problem.

The losses in an ordinary reciprocating pump may be classified as follows:-

1.- The mechanical losses of the pump itself due to friction in the gears, bearings, stuffing boxes and plunger packing.

2.- Leakage losses through the closed valves and around the plunger.

3.- Water frictional and eddy losses in the passages of the pump.

4.- Losses due to accelerating and retarding the water columns.

5.- Losses in the suction and discharge valves due

to restricted passages, sudden turns and to the rushing back of the water before closure.

Most of the losses have been pretty carefully studied, and the laws governing them under varying conditions established as well as the proper range they should have under good operating conditions.

While current practice has approximately fixed the value of the valve losses, this is the result largely of practical experience and the laws governing the action of the valves under various conditions are but imperfectly understood.

The experiments herein described were undertaken primarily to study the laws governing the valve motion to the end that its action under any given set of conditions could be better forseen. In addition the scope of the problem was made to include an analysis of the various pump losses and a comparison of these under the different operating conditions.

It was thought best to use the pump without alterations other than those found necessary to put it in good running condition. While this has in some cases obscured to a degree the law of action of the valves, it has better enabled us to approach the true conditions existing in practice.

THEORY

We have not been able to find an adequate discussion of the theory of valve action in English. Neither have we found published reports of tests of pump valves, except some three or four articles refered to in the bibliography that are quite limited in the field they cover. A number of articles deal with special forms of valves which the designers advocated or used from considerations of experience in pump operation.

German and French technical literature contains a number of articles dealing with the theory of valve action and also others describing tests of pump valves in operation.

The following discussion is taken largely from a discussion by H. Berg. (See "Die Pumpen" 3rd edition or "Zeitshrift des Vereines Deutcher Ingenieure", Vol. 48, pg. 1093 July 1904)

Theory of Valve Action for Crank and Flywheel Pumps

Fundamental Considerations

We will consider the discharge valve of a single acting pump.

Let F = Area of piston .

u = Velocity of piston .

f_1 = Area of the free opening of the valve seat

c_1 = Velocity of water through valve seat

Then

(1) $f_1 c_1 = F u$, or $c_1 = \frac{Fu}{f_1}$

The velocity of water in the valve seat is thus proportional to the velocity of the piston and increases from zero to a maximum and back to zero in unison with the piston velocity. Corresponding to the variation of the velocity of the water the valve is lifted from its seat, rises to a certain height and then drops back again on its seat. The law of the valve motion is thus based on the law of the piston motion.

Before ascertaining the law of valve motion more accurately it will be well to learn the general character of valve motion by means of simple formulae neglecting certain features. For this purpose we will neglect the displacement of the valve itself throughout a discharge stroke and assume furthermore that it floats at rest on the stream of water. Then in each moment the same quantity of water is discharged through the valve opening as is discharged through the valve seat.

Let h = valve lift

l = circumference of the valve

c = velocity of the water discharging through the valve lift opening.

a = coefficient of contraction, which is variable with the lift but is assumed independent of the velocity of efflux.

From the foregoing the following relation holds

(2) $a\ c\ l\ h = f_1 c_1$

or with equation (1)

(3) $a\ c\ l\ h = F\ u$

which means valve discharge is equal to piston displacement.

Transposing

(4) $h = \frac{F\ u}{a\ c\ l}$

Considering for the present a connecting rod of infinite length, the piston velocity

(5) $u = w\ r \sin \beta$

where $w = \frac{2\pi n}{60} = \frac{\pi n}{30}$

n = Revolutions per minute

r = The crank radius

β = Angle the crank makes with its dead center position.

Combining equations (4) and (5) we obtain

(6) $h = \frac{F\ r\ w}{a\ c\ l} \sin \beta$

From which it follows, if the coefficient a is constant and the velocity c does not vary, that the valve lift is proportional to the sine of the crank angle.

When $\beta = 0°$ or $180°$ then $h = 0$

When $\beta = 90°$ $\sin \beta = 1$ its maximum value from which $h_{max} = \frac{F\ r\ w}{a\ c\ l}$

If we plot the crank angle as abscissae and the corresponding valve lift as ordinates, we obtain a sine curve for the valve lift diagram. (See fig. 1)

VALVE LIFT DIAGRAM.

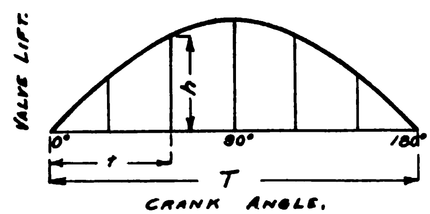

FIG. 1.

As the crank turns with uniform velocity, equal crank angles are turned in equal intervals of time. The base line of the figure represents a crank angle of 180° and also represents the time of the piston stroke. The abscissae of any point of the valve lift line represents the time t of a certain valve lift h reckoned from the beginning of the piston stroke.

In the infinitesimal time dt the position of the valve, with respect to its seat, changes by dh, the velocity of the valve $v = \frac{dh}{dt} = \tan \theta$ where θ is the angle the tangent to the curve makes with the base line.

As is seen by the figure, the valve rises with a constantly increasing velocity until at the maximum lift the velocity becomes zero and then descends again according to the same law but with constantly increasing velocity. By means of equation (6) we get the velocity

$$v = \frac{dh}{dt} = \frac{Frw}{a cl} \cos \beta \frac{d\beta}{dt}$$

or as $\frac{d\beta}{dt} = w$

$$(7) \quad v = \frac{Frw^2}{a cl} \cos \beta$$

The valve velocity is hence proportional to the cosine of the crank angle.

For $\beta = 0°$ that is $\cos \beta = 1$; $v = \frac{Frw^2}{a cl}$

" $\beta = 180°$ " " " $\beta = -1$; $v = -\frac{Frw^2}{a cl}$

" $\beta = 90°$ " " " $\beta = 0$; $v = 0$

For all angles from 0° to 90° the cosine is positive and v is also positive, or $v = +\frac{dh}{dt}$ The valve lift increases and the valve rises.

For all angles from 90° to 180° the cosine is negative and v also is negative, or $v = -\frac{dh}{dt}$ The valve lift decreases and the valve descends.

Plotting the values of β and v we get the velocity diagram which is a cosine curve. (See fig. 2)

VALVE VELOCITY CURVE

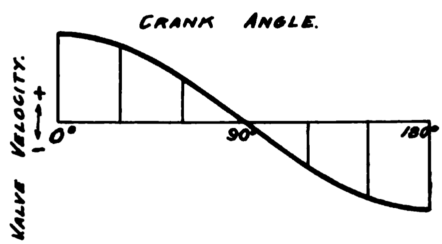

FIG. 2.

Velocity is the greatest as the valve leaves its seat and again as it returns to the seat. It is zero at the maximum lift point.

The change in the valve velocity (acceleration or retardation) may be expressed by

$$k = \frac{dv}{dt}$$

and from equation (7)

$$k = \frac{dv}{dt} = -\frac{Frw^2}{acl} \sin\beta \frac{d\beta}{dt}$$

$$(8) \quad k = -\frac{Frw^3}{acl} \sin\beta$$

For $\beta = 0°$ or $180°$; $\sin\beta = 0$, so that $k = 0$

" $\beta = 90°$; $\sin\beta$ has its maximum value, that is

$$\sin\beta = 1 \text{ and hence}$$

$$k = -\frac{Frw^3}{acl}$$

From equation (8) it should be noted that all values of the valve lift acceleration are negative.

The velocity of valve lift, as found above, is positive from 0° to 90° and since the acceleration is negative the motion of the valve is retarded in its ascent. From 90° to 180° the velocity of the valve is negative and the acceleration being also negative, the motion of the valve is accelerated in its descent.

The greatest change of velocity takes place at the reversal of motion of the valve, that is, at the highest

valve lift. At the beginning and end of the valve discharge the change of velocity is zero.

Fig. 3, being a sine curve, is a graphical representation of equation (8)

ACCELERATION DIAGRAM.

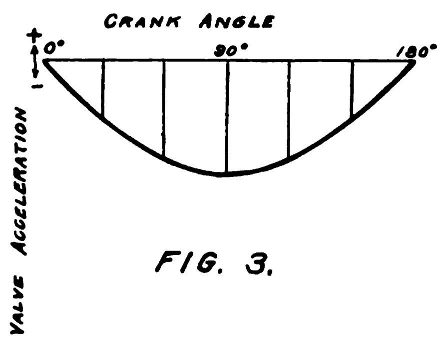

FIG. 3.

From equations (6) and (8) we obtain

(9) $k = -w^2h$

The acceleration of the valve is thus proportional to the valve lift.

Laws of Motion of the Valve

We will now consider the motion of the valve or the effect of its displacement. The volume of discharge through the valve opening is no longer equal to the discharge through the seat at any time, or

$aclh = f_1c_1$ no longer holds.

This is seen when one remembers that a volume of water equal to a column the diameter of the valve and the height of the lift passes through the seat, as the valve ascends, which does not pass through the valve opening. As the valve descends to its seat this column of water is forced out through the valve opening in addition to the water coming through the seat at the time. This will cause the valve lift to be less than is expected from the piston velocities the first half of the stroke and greater than expected the last half of the stroke.

Let f = The area of the valve

v = " velocity of the valve

Then equation (2) can be altered thus to include this effect just mentioned

$$(10)\quad a c l h = f_1 c_1 - f v$$

The volume discharged through the lift opening = volume of water discharged through the valve seat - valve displacement, for which "v" is positive for the ascent and negative for the descent of the valve. (See fig. 2)

Since from equation (1)

$$f_1 c_1 = F u$$

the following equation also holds

(11) $aclh = Fu - fv$

Volume through the lift = piston displacement - valve displacement. From this the valve lift is fixed by

(12) $$h = \frac{1}{acl}(Fu - fv)$$

or as the velocity of the valve $v = \frac{dh}{dt}$

$$h = \frac{1}{acl}\left(Fu - f\frac{dh}{dt}\right)$$

The integration of this differential equation is possible if a and c are assumed constant. The accomplishment of this difficult task leads to an equation which requires another assumption to put it into practical form. This ultimate result can be obtained in another way.

For the determination of the valve lift

$$h = \frac{1}{acl}(Fu - fv)$$

it is necessary to know v, the velocity of the valve. This may be obtained by conversion of the equation

(13) $$v = \frac{dh}{dt} = \frac{1}{acl}\left(F\frac{du}{dt} - f\frac{dv}{dt}\right)$$

in which a and c are assumed constant.

The expression in the parenthesis $\left(F\frac{du}{dt} - f\frac{dv}{dt}\right)$ may be expressed in words, as piston area times piston acceleration minus valve area times valve acceleration. The valve acceleration, $\frac{dv}{dt}$, is however small. It is further-

more, as is shown in fig. 3, smaller as the valve approaches its seat and becomes, at the closing of the valve, infinitely small.

As it will be sufficient for practical needs if the calculation considers the conditions at valve closure, it seems reasonable to neglect the term $f \frac{dv}{dt}$ in equation (13) obtaining

$$(14)\quad v = \frac{F}{acl}\,\frac{du}{dt}$$

Inserting this value in equation (12)

$$(15)\quad h = \frac{1}{acl}\left(Fu - \frac{fF}{acl}\,\frac{du}{dt}\right)$$

which gives the value of lift dependent on the piston velocity.

For a connecting rod of infinite length $u = rw \sin\beta$

$$\frac{du}{dt} = rw\cos\beta\,\frac{d\beta}{dt} = rw^2\cos\beta$$

Substituting these values in equation (15)

$$(16)\quad h = \frac{1}{acl}\left(Frw\sin\beta - \frac{Frw^2 f}{acl}\cos\beta\right)$$

$$(17)\quad \text{or}\quad h = \frac{Frw}{acl}\left(\sin\beta - \frac{fw}{acl}\cos\beta\right)$$

An equation for the valve lift of a pump with crank and flywheel drive.

Multiplying both terms of equation (16) by acl there results

$$(18)\quad aclh = Frw\sin\beta - \frac{Frw^2 f}{acl}\cos\beta$$

The volume discharged through the lift opening = the piston displacement minus the valve displacement.

In fig. 4 is shown graphically the relation between the volume discharged through the valve opening, and the valve lift.

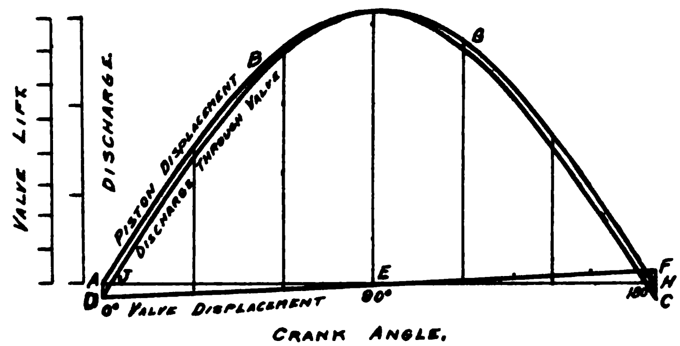

FIG. 4.

The sinous line ABC shows the piston displacement, and the valve displacement is shown by the cosinous line DEF. Adding the ordinates of these two lines algebraically there results the graphical expression of equation (18), the line DGH showing the volume discharged through the valve lift.

The latter line however represents at the same time the valve lift, as one obtains equation (13) for the valve lift by dividing equation (18) by acl. For the representation of the valve lift one should divide the

ordinates of fig. 4 by acl to obtain the valve lift.

The valve lift line now shows the following important facts. The valve first begins to open after the piston begins to make its return stroke and has traversed a certain distance AJ from the dead center. The greatest lift is reached after the crank has passed the 90° point, which would be even more pronounced with a finite connecting rod. The valve is still open a small amount at the time of the reversal of the piston that is at a crank angle of 180° and closes a short time after the piston has begun its return stroke.

The fact that the discharge valve does not open simultaneously with the beginning of the discharge stroke is understood better when one considers that the suction valve, which acts in the same manner as the discharge valve, is not on its seat at the beginning of the discharge stroke and that the discharge valve will not leave its seat until the suction valve is closed. The opening of the discharge valve is therefore dependent on the time of closing of the suction valve.

Equations for Computing the Discharge Through the Valve.

Equation (17), $h = \frac{Frw}{acl} \left(\sin\beta - \frac{fw}{acl}\cos\beta\right)$ gives the valve lift dependent on the velocity through the

valve lift area. As in most practical cases not the velocity through the valve lift but the valve loading producing this velocity is given, we must formulate an equation for the valve lift dependent on the effect of the valve loading.

NOTE--- We have used the term "valve loading" to designate the total load tending to force it toward the seat in any position and includes both the effect of the spring and the action of gravity on the valve and spring mass.

The relation between the valve loading and the velocity through the valve lift may be derived by means of the experiments made by C. von Bach (C. von Bach: "Versuche über Ventilbelastung und Ventilwiderstand, Berlin 1884, Julius Springer. Versuche zur Klarstellung selbsttätiger Pumpenventile, Z. 1886 S.421 w.f.) with nine different designs of valves and also the experiments made by H. Berg on saucer shaped valves.

For a stream of water of uniform depth discharging through the opening between the valve seat and floating valve, the relation, as given by Bach, between the force P_1 exercised by the stream on the lower side of the valve and the velocity c_2 with which the water approaches the bottom of the valve, is given by an equation of the following form.

$$\frac{P_1}{f_1 Y} = Z_1 \frac{c_2^2}{2g}$$

in which

f_1 = Area of the valve seat opening.

Y = Weight of one cubic *unit* of water.

g = Acceleration due to gravity.

Z_1 = A coefficient which is dependent on the design of the valve and varies with the lift.

If the water has a velocity c_1 through the valve seat and if the valve moves with a velocity of v, the velocity with which the water impinges against the valve is equal to $(c_1 \mp v)$ in which the upper sign holds for the rising valve and the lower sign for the descending valve. From this the force exercised by a stream of water on a moving valve is determined by

$$\frac{P_1}{f_1 Y} = Z_1 \frac{(c_1 \mp v)^2}{2g}$$

This equation shows that for a certain valve lift there is a certain value of Z_1, and that the force of the stream of water is proportional to the square of the velocity with which the water impinges against the valve.

But now the discharge through the valve lift is determined by equation (10) by means of

$$a c l h = f_1 c_1 \mp f v$$

$$f_1 c_1 \mp f v \text{ is approximately } = f_1 \, (c_1 \mp v)$$

and from this we get the relation

$$\frac{c}{c_1 \mp v} = \frac{f_1}{alh}$$

The velocity of discharge c through the valve lift opening is approximately proportional to the velocity $(c_1 \mp v)$.

From this the following equation holds $\frac{P_1}{f_1 Y} = Z_2 \frac{c^2}{2g}$

or if we introduce the area f of the valve we get

$$(19) \quad \frac{P_1}{fY} = Z \frac{c^2}{2g} \; ; \quad c = \frac{1}{\sqrt{Z}} \sqrt{2g \frac{P_1}{fY}}$$

Let W = Weight of the valve in water.

S = The spring tension or load due to the spring.

M = Mass of the valve.

k_1 = Acceleration of the valve.

The forces acting on the valve are, the upward pressure P_1 of the stream of water and the downward pressure of the valve loading (W + S). The difference $P_1 - (W + S)$ is the force which accelerates the valve. From this

$$Mk_1 = P_1 - (W + S)$$

or $P_1 = W + S + Mk_1$

$(W + S + Mk_1)$ is the effective valve loading which is made up of the actual valve load (W + S) and the mass force Mk_1.

Substituting in equation (19)

$$\frac{W + S + Mk_1}{f\,Y} = Z\,\frac{c^2}{2g}$$

The effective load is thus proportional to the square of the velocity through the valve opening and this is determined by

$$c = \frac{1}{\sqrt{Z}}\sqrt{2g\,\frac{W + S + Mk_1}{f\,Y}}$$

Assuming the effective valve load to be replaced by a water column of area = f, and height = b, and putting $W + S + Mk_1 = fbY$ we get

$$c = \frac{1}{\sqrt{Z}}\sqrt{2gb}$$

in which

$$(20)\quad b = \frac{W + S + Mk_1}{fY}$$

giving the effective valve load

Neglecting the mass force, this equation holds

$$(21)\quad b = \frac{W + S}{fY}$$

Since $c = \frac{1}{\sqrt{Z}}\sqrt{2gb}$ we can now substitute for ac the expression $\frac{a}{\sqrt{Z}}\sqrt{2gb}$ or designating $\frac{a}{\sqrt{Z}}$ by m

$$(22)\quad ac = m\sqrt{2gb}$$

This coefficient m, also a and Z from which it is composed, is dependent on the valve lift; it takes

into consideration the contraction in the valve lift area and the relation between valve load and the velocity through the valve lift.

In order to obtain the equation for computing the valve discharge we now get, if we assume m and $\sqrt{2\,gb}$ invariable as the values of a and c were assumed above, the following:-

A-- Without taking into consideration the finite length of the connecting rod.

From equations (17) and (22) the valve lift is;

$$(23)\quad h = \frac{Frw}{ml\sqrt{2\,gb}}\left(\sin\beta - \frac{fw}{ml\sqrt{2\,gb}}\cos\beta\right)$$

Let the crank angle at closure equal

$$\beta = 180^\circ + d^\circ$$

This angle may be obtained by putting h = 0 in the previous equation.

Then

$$0 = \frac{Frw}{ml\sqrt{2\,gb}}\left[\sin(180 + d) - \frac{fw}{ml\sqrt{2\,gb}}\cos(180 + d)\right]$$

also

$$(25)\quad \tan(180 + d) = \tan d = \frac{fw}{ml\sqrt{2\,gb}}$$

The angle d will be named the angle of lag.

To determine the maximum valve lift:

The maximum valve lift differs so little from the valve lift at 90° crank angle, that for all practical

purposes, the valve lift at 90° crank angle may be taken as the maximum.

With this we obtain from equation (23)

$$h_{max.} = \frac{Frw}{ml\sqrt{2\ gb}}\left(\sin 90 - \frac{fw}{ml\sqrt{2\ gb}}\cos 90\right)$$

$$(25)\quad h_{max.} = \frac{Frw}{ml\sqrt{2\ gb}}$$

To determine h_o the valve lift at the reversal of the piston. For $\beta = 180°$ equation (23) gives the following:-

$$h_o = \frac{Frw}{ml\sqrt{2\ gb}}\left(\sin 180 - \frac{fw}{ml\sqrt{2\ gb}}\cos 180\right)$$

$$(26)\quad h_o = \frac{Frw^2 f}{(ml\sqrt{2gb})^2}$$

To determine v_1 the velocity with which the valve closes. From equations (11) and (22)

$$mlh\sqrt{2\ gb} = Fu - fv$$

At valve closure $h = 0$ and $\beta = 180° + d$

So that $u = rw \sin (180 + d) = - rw \sin d$

From this

$$0 = - Frw \sin d - fv_1$$

$$(27)\quad v_1 = - \frac{Frw}{f}\sin d$$

B---- Considering the finite length of the connecting rod.

Let

L = the length of the connecting rod.

The piston velocity is now given by

$$u = rw \left(\sin\beta \pm 1/2 \frac{r}{L} \sin 2\beta\right)$$

in which the upper sign is used for the forward stroke and the lower sign for the return stroke of the piston. The acceleration of the piston is then given by

$$\frac{du}{dt} = rw \left(\cos\beta \frac{d\beta}{dt} \pm \frac{r}{L} \cos 2\beta \frac{d\beta}{dt}\right)$$

$$= rw^2 \left(\cos\beta \pm \frac{r}{L} \cos 2\beta\right)$$

Then from equations (15) and (22) the valve lift is obtained

$$(28)\quad h = \frac{Frw}{ml\sqrt{2gb}} \left[\left(\sin\beta \pm 1/2 \frac{r}{L} \sin 2\beta\right) - \frac{fw}{ml\sqrt{2gb}} \left(\cos\beta \pm \frac{r}{L} \cos 2\beta\right)\right]$$

To determine the angle of lag d:

With $h = 0$ and $\beta = 180° + d$ there results

$$0 = \text{sine}\,(180 + d) \pm 1/2 \frac{r}{L} \sin(360 + 2d) - \frac{fw}{ml\sqrt{2gb}} \left(\cos(180 + d) \pm \frac{r}{L} \cos(360 + 2d)\right)$$

that is

$$\frac{-\sin d \pm 1/2 \frac{r}{L} \sin 2d}{-\cos d \pm \frac{r}{L} \cos 2d} = \frac{fw}{ml\sqrt{2gb}}$$

for which we can put

$$(29)\quad \tan d = \frac{fw}{ml\sqrt{2\,gb}}$$

To determine the maximum valve lift h_{max} :

$$h_{max} = \frac{Frw}{ml\sqrt{2gb}}\left(1 \pm \frac{fw}{ml\sqrt{2gb}}\,\frac{r}{L}\right)$$

$$(30) = (\text{approx})\ \frac{Frw}{ml\sqrt{2\,gb}}$$

To determine the valve lift at dead center h_o:

Let

$\beta = 180°$ and from equation (28) we have

$$h_o = \frac{Frw^2}{ml\sqrt{2\,gb}}\left[\left(\sin 180 \pm 1/2\,\frac{r}{L}\,\sin 360\right) - \frac{fw}{ml\sqrt{2\,gb}}\left(\cos 180 \pm \frac{r}{L}\,\cos 360\right)\right]$$

$$(31)\quad h_o = \frac{Frw^2}{(ml\sqrt{2gb})^2}\left(1 \mp \frac{r}{L}\right)$$

To determine the final velocity v_1 with $h = 0$ and

$$u = rw\left[\sin\,(180 + d) \pm 1/2\,\frac{r}{L}\,\sin\,(360 + 2d)\right]$$

From equation (11)

$$0 = Frw\left(-\sin d \pm 1/2\,\frac{r}{L}\,\sin 2d\right) - fv$$

$$(32)\quad v_1 = -\frac{Frw}{f}\left(\sin d \mp 1/2\,\frac{r}{L}\,\sin 2\,d\right)$$

Computing the Valve Lift Line Without Taking The Mass of the Valve into Consideration:

The development of the equation for valve lift is based on the assumption that the coefficient of con-

traction a and the velocity c through the valve lift, also that the coefficient m and the valve load b have constant values during the entire motion of the valve. Actually these values vary with the valve lift. Equations (23) and (28) therefore only hold for constant values of m and b. But for any given valve lift h, the values m and b may be assumed constant and for the short interval of time in which the valve is at the given lift the relations of equations (23) and (28) hold. If the values which m and b have for the different valve lifts are known, successive values of h may be taken and the corresponding crank angle β computed from equations (23) and (28). This gives any desired number of points for plotting the valve lift diagram..

The connection between h and b is solved in the following way. Let C = the spring constant, that is the pressure which the spring exerts for each unit of compression in length; Y_0 the total length the spring is compressed when the valve is on its seat and the spring is in position over it; h equal, as before, the valve lift.

Then

(33) $S = (Y_0 + h)\ C$

The relation between the valve load b and the valve

lift h is then determined from equation (21)

$$(34)\quad b = \frac{W + (Y_0 + h)C}{f\,Y}$$

The connection between the coefficient m and the valve lift h is conditioned on the construction of the valve and can only be determined experimentally.

PUMP LOSSES

The losses of power in the pump may be divided into two classes, viz: mechanical losses and hydraulic losses. The former consists of gear and journal friction and the friction due to the packing of the piston rods and plungers. The piston rod and plunger friction depends upon the condition of the packing and the lubrication of same, which is subject to large variations. The hydraulic losses include those due to pressure head losses through the valves, frictional and eddy losses in other passages of the pump, and the losses due to leakage and slip. The indicated horse power obtained from the plunger chamber card represents the energy exerted upon the water, and hence includes the power necessary to overcome all the hydraulic losses. But the mechanical losses are absorbed in the mechanical parts of the pump and therefore are not included in indicated horse power. That is, the power input minus the indicated horse power is the energy lost in mechanical friction. The power output of the pump is less than the indicated horse power by the amount of the hydraulic losses. The power lost due

to the valves may be separated from the hydraulic losses if the difference in pressure through the valves is known. This may be determined from the indicator cards.

Many hydraulic losses are supposed to vary nearly as the square of the velocity. Take, for instance, the loss in the valve seat alone. This loss is very much like the loss of head through a submerged orifice. Further, some hydraulic losses such as leakage and slip are increased by an increase of pressure as well as by an increase of speed.

It is shown in works on hydraulics (see "Treatise on Hydraulics" by Mansfield Merriman, p. 134, 8th edition) that the loss of head h^1 when a liquid flows through an orifice, tube or pipe is

$$h^1 = \left(\frac{1}{C^2} - 1\right)\frac{v^2}{2g}$$

where v = theoretical velocity of the water due to the impressed head, and C = the coefficient of velocity.

Let W = the weight of water passing per minute then

$$Wh^1 = \left(\frac{1}{C^2 - 1}\right)\frac{Wv^2}{2g}$$

equals the energy lost per minute in the given passage. As C does not vary between very wide limits for

ordinary conditions met in practice, we may say the loss will vary nearly as the square of the velocity.

It follows, then, that we should expect the losses through the valve seats and other passages of the pump to vary approximately according to this law.

This, of course, does not apply to leakage and probably slip which depend rather on the difference between the discharge and suction pressures. The quantity of leakage from $q = c\ a \sqrt{2gh}$ must vary nearly as the square root of the head.

DESCRIPTION OF APPARATUS

Motor

The source of power used in these experiments was a Northern Electric Co. 25 H. P., variable speed, (600-1050 R. P. M.), 500 volt, 43 ampere, D. C. Shunt wound motor hung from the ceiling. By means of a field rheostat combined with the starting box, the speed of the motor could be varied so as to give the crank of the pump almost any speed from 35-70 R. P. M. Motor pulley was 16 in. in diameter.

Power Transmission

The power was transmitted by means of an 8 in. Gandy belt from the motor to a line-shaft, 26 ft. long, having six ordinary hanger bearings. The power was transmitted from the line-shaft to the jack-shaft of the pump by means of an 8 in. leather belt. The distance between motor and line-shaft was 19 ft. and the distance between the line-shaft and the jack-shaft was 19 ft. also. The diameters of pulleys on line-shaft were 16 in. and the diameters of the tight and loose pulleys on jack-shaft were 40 in.

Pump

(See Plates A and 1)

The pump tested was a Fairbanks-Morse duplex, double-acting, outside center-packed, crank-driven, reciprocating plunger pump.

The pump was of special design in that a 6 in. plunger was substituted for the 7 in. for which the rest of the pump was designed. Thus the ratio of valve area to plunger area was correspondingly increased over the current practice of the builders.

The stroke was 10 in., suction pipe 6 in. with open end, and discharge pipe was 5 in. The jack-shaft was geared to the crank. In these experiments the pump was connected in the two ways as shown by the full and dotted lines on drawing, Plate I. The pump was first "short circuited" in order to get the valve losses and net power, as near as possible, to circulate the water thru the pump, and general behavior of valves with the suction under pressure. With the connections as shown by dotted lines water was taken from a basin directly below the pump, at a constant distance of 6 ft. below the center line of the suction pipe, and discharged over a calibrated 6 in. trapezoidal weir, slope 1 : 2, back to the basin. The depth of water in the pit was obtained from a float gage, the float and rod of which were placed in a wooden box to eliminate the effects of whirls and

eddies in the basin. The rod was read on a scale placed vertically by the side of the groove which guided the rod from the float.

The ratio of the connecting rod to the crank was 6:1, the former being 30 in. and the latter 5 in. The piston rods were of 1-3/4 in. diameter thus making the crank end displacement during one stroke 258.69 cu. in. while the head end displacement was 282.74 cu. in. The pump was equipped with a revolution counter actuated by the cross head thru a reducing motion.

Valves

In each deck, both discharge and suction, were 3 "saucer-shaped," spring loaded valves, having a maximum lift of 7/8 in. up to the stop. The studs screwed into the radial ribs at the center of the seat. The valve details are shown on Plate 4 & 5 (see also Plate B). The arrangement of the valves in the decks is shown diagramatically by Plate 2.

Gages

Two ordinary Bourdon gages were used on the discharge while a mercury manometer was used on the suction pipe just before it entered the pump. One pressure gage was used for low and one for high pressures. In the short circuit runs an open water tube was used

to measure the smallest heads.

Electrical Instruments

The power input to the motor was measured by means of the portable types of Weston Standard Voltmeter and Ammeter. Previous experimenters found errors in voltmeter readings due to the influence of the large currents in the ammeter circuit when the instruments were close together and hence these instruments were placed several feet apart.

The electrical connections to the apparatus are shown on Plate 3 M is the motor driving the drums in the photographic apparatus, A is the arc light, E_c and E_t are the cross head and time electromagnets respectively on the indicators, while E_{pc} and E_{pt} are the corresponding magnets on the photographic apparatus, C is a condenser, L is a lamp bank, G is a ground, R is a relay and B is a battery.

Indicators

(See Plate B for arrangement.)

In all 5 indicators were used. On the pump proper there was one below the suction deck, one in the plunger chamber between decks and one immediately above the discharge deck. The last two were the outside spring type of Crosby hydraulic indicators while

the other was the inside spring style. In order to more clearly obtain the true pressures at the respective points these three indicators were connected directly by brass sleeves, without cocks, as closely to the pump as possible and the passage thru the sleeves had as large an area as the piston of the indicators. The drums of these three indicators were driven by cords from a common jack-shaft run by a small hydraulic motor operating from the city pressure. Any desired speed of the drums was obtained by throttling the motor. The paper on the drums was held by rubber bands. (See photo, Plate B)

Attached to each of these indicators were two electro-magnets, (See Plate B) which marked seconds in time and the dead centers of the pump piston travel upon the paper. The time was obtained by a contact on the pendulum of the clock in the Physics Department and the dead centers were obtained by a contact on the cross head of the pump. The electrical arrangement is shown on plate 3.

Two other indicators were used - one on the discharge pipe proper and one on the suction. The drums of these two were given a reciprocating motion by connecting them to a reducing mechanism operated directly

by the cross head,- as is generally done in steam engine testing. These were inside spring indicators and were connected to the pipes in the usual manner.

Photographic Valve Action Recording Apparatus.

To each valve was attached a universal joint as is shown in Plate 4 and B. . A small brass rod screwed into the upper stirrup and could be adjusted by means of a lock nut. At the upper end of the brass rod a small aluminum target, having a small hole drilled through it, was screwed on. The brass stem communicated the motion of the valve to the target. The target holder or case consisted of a small brass box with two detachable sides, being held by screws. These two sides had small vertical slits and directly back of each face was a thin sheet of mica or celluloid. The aluminum target merely moved in the holder, which was screwed into the cap of the discharge chamber. The stem worked loosely enough thru the holder to permit the full pressure in the chamber to come up around the target. There was no friction along the stem or target save the skin friction of the surrounding water. Any entrapped air could be let out by loosening the screws in the face plates.

The photographic part of the apparatus consisted essentially of a small arc light, a couple of electro-

magnets, two drums upon which sensitised paper could be wound and drawn from one side to the other, all of which was contained in a light-proof brass box. Upon top of the box a small D. C. electric motor was pivoted and could be thrown in gear to turn the vertical shaft of one of the drums inside the box.

In the bottom of the box a slot was made so that almost the entire target holder could project into the box. When the apparatus was thus placed over any target holder the only light which could come to the sensitised paper was thru the small hole in the aluminum target. By means of a shutter this too could be shut off. The electromagnets merely operated two separate shutters which would allow two more pencils of light to pass to the sensitised paper. These magnets were operated to mark the dead centers of the stroke and seconds of time and were connected in series with the corresponding magnets of the indicators already described. The marks on the paper were above and below any possible trace of the target motion. Thus when operating, the motor drew the paper past the target which was moving vertically with the valve. The electromagnet marks merely coordinate the valve motion with respect to time and position of piston.

The slot thru which the target holder projected was equipped with a spring flap which kept light out when the instrument was not in position on the pump. One side of the box could be slipped off for taking out or putting in the paper. A small notched wheel on the drum shaft was operated by an eccentric and ratchet arrangement so as to give the amount of unexposed paper on the drum.

Prony Brake.

The power input to the pump was obtained with a Prony brake. The brake wheel was attached permanently to the loose pulley on the jack-shaft of the pump. By simply shifting the belt from the tight to the loose pulley and placing the brake band on the brake wheel, brake runs could be made to obtain the power delivered at the jack-shaft for any condition of switchboard readings. We were thus able to check the power input to the pump just at the time of taking a run. The brake arm was 3.49 ft. in length. The brake wheel differed from the ordinary brake wheel by having a flange on each edge of the rim to hold the brake band in place. By using these flanges instead of the lugs on the brake band itself, which are ordinarily used, any water which splashed onto the wheel was thrown out radially and

off the wheel instead of working under the brake band. The opening in the brake band was also placed at the top and at a point where all the slack would be drawn, making it easier and more convenient to manipulate.

Calibrations

All instruments and apparatus were calibrated and corrections were applied in all data worked up. The indicators and pressure gages were calibrated by means of the Crosby Weight Gage Tester. Both ammeter and voltmeter were calibrated by comparison with the Laboratory Standards in the Electrical Engineering Department. The scales used in the brake runs were tested with the Laboratory Standard Masses.

The accuracy of reproducing the true motion of the valve on the sensitised paper was determined by making a special target, having several holes 1/8 in. apart vertically. By measuring the card when developed it was found that the spacing on the cards was slightly magnified due to the target being between the light and the paper, the rays of the arc light not being parallel. The box sometimes had to be placed with different faces of the target holder (on different targets) toward the light and hence the true values of lift were obtained by multiplying the card values by either 93% or 95.5%.

All valve springs were calibrated up to their maximum compression by the apparatus shown on Plate 6. Sand was poured into the bucket and the load on the spring was obtained for each amount of compression as shown by the notch along the scale. Thus the initial tension of the spring as placed in the valve chamber could be determined from its measured length under those conditions. The weights of the valves, with and without the attachments, were measured. The specific gravity was also determined in order to be able to know the effective weight of the valve submerged such as would be the case when the valve was operating in the chamber full of water.

METHOD OF EXPERIMENTING

Preliminary Runs

For the purpose of preliminary runs the pump was short-circuited by connecting the discharge to the suction, as shown on Plate / . It was desired to determine the amount of power required to operate the pump, and to circulate the water through the passages and valves. Runs were made at five speeds corresponding to five points on the regulating rheostat from lowest to highest. These were designated A, B, C, D, E, corresponding respectively to positions of the speed regulating lever on its contact points, 1, 5, 10, 15, 21. Revolutions per minute of the pump varied with the load and voltage, and were 36.0 to 41.8, 40.7 to 44.3, 47.9 to 51.8, 55.6 to 60.3, 61.4 to 69.3 respectively. Runs were first taken with each speed and with all the valves operating. One valve in each deck was then blocked down and another set of runs was taken. These runs were designated A_1, B_1, C_1, D_1, E_1. Finally, to study the effect of greatly restricted valve area, two valves in each chamber were blocked, leaving only one valve to operate. These runs were designated A_2, B_2, C_2, D_2, E_2. Before each run was

taken, time was given for conditions to become steady.

With each run indicator cards were taken every ten minutes, making four sets per run. In nearly all cases cards were taken on each of the three indicators on the pump, and on those on the suction and discharge pipes. The cards from the plunger and discharge chambers were made simultaneously and the others as soon thereafter as possible. Cards for the atmospheric pressure line were taken on the three pump indicators as often as necessary, or as thought advisable. Photographic records were obtained for each discharge valve except D_2 and D_3, which were not accessible for taking records. In addition, the maximum lift for each discharge valve was measured with a scale. One set of records and measurements was taken with each run.

In the short-circuit or circulating runs the regular data was all taken first, then the brake data was taken for the several speeds used in these runs. But from rating curves previously obtained for the motor and shafting it was found that the losses of power in this part of the apparatus were quite variable. It was evident then that this method would not give good results, so in all other runs the brake data was taken immediately after the pump run. More will be said

concerning the accuracy of these rating curves under "Motor Rating."

Regular Runs

By the regular runs are meant those taken with the pump connected so as to draw from the suction pit and discharge over the weir. In addition to the data taken in the preliminary runs, the hook gage was read every two and one-half minutes, and indicator and plunger leakage was measured.

With each of the five speeds five different discharge pressures were taken, 15, 30, 45, 60, 75 pounds. It should be said here that variations in the speed of the pump, caused by voltage fluctuations tended to vary the discharge pressure. But this pressure was held constant by careful regulation of the discharge gate valve. The suction head was read on the mercury manometer and was kept nearly constant with the float gage. The first of the regular runs were performed with only a part of the valves operating, the others being blocked down. But it was noticed on the indicator cards, by the rounding of the lower corners and the slow rise and fall of pressure in the plunger chamber, that air was being drawn into the pump. This effect is shown on the indicator card in Plate 7 . It was

decided that air entering through the suction indicators and possible leaks in the suction line was entrapped under the valves blocked down. So these runs were abandoned and in the remaining tests the valves were all permitted to operate. The effects mentioned above were no longer noticeable.

The brake data with a few exceptions was taken immediately after the pump run. Usually three brake loads were used, the scales being balanced to read the net load. Such values were used as would give ammeter readings above and below, and one about equal to that observed while the pump was running. For each brake load at least two speed observations were made, each of one minute's length. If these two values did not check within two percent, one or two more observations were made. Five readings of the voltmeter and ammeter were taken each minute, or during each speed determination.

Two sets of photographic cards were obtained on each run, and in each set were found usually two lift cards for each valve.

A sample log sheet is reproduced and shown in Plate 17

Auxiliary Experiments

1. Pressure to Lift Valve.

In a saucer-shaped valve a portion of the area of

the lower side is ground to the seat to form contact area. Then the remaining surface not so used is the area exposed to water pressure. And theoretically if the valve seats perfectly, a unit pressure must be exerted upon the lower side greater than that upon the upper side by an amount sufficient to make the forces on the two sides equal in order that the valve may lift. To overcome the weight of the valve and the spring tension, the lower force must exceed the upper.

If P_u and P_l = pressures per square in. on upper and lower sides of valve respectively,

F_u and F_l = the respective total forces due to these pressures,

A_u = area of upper side of valve,

A_l = area of lower side of valve,

A_c = contact area of valve,

W = weight of valve and tension of spring

Then $A_l = A_u - A_c$, and for the valve to rise

$F_l - W = F_u$, or $P_l\,(A_u - A_c) = P_u A_u + W$,

Or

$$P_l = \frac{P_u A_u + W}{A_u - A_c}$$

that is, the pressure under the valve must exceed that above depending upon the contact area of the seat and the weight and spring tension. This means that the valve would jump from its seat with shock, for at the

instant unseating begins the higher pressure below becomes effective over the contact area and exerts force over the entire lower side of the valve. However, valves do not seat perfectly, and the less the contact area which is effective or absolutely tight, the less will be the effect described above.

In order to study this effect, chamber A was fitted up with water columns connected, one on each side of the valve deck. Water from the mains was tapped into the plunger chamber through a valve. By admitting water and observing the water columns the difference in pressure across the valve was directly measured. Several trials were made on each valve in the discharge deck, the pump being idle of course. In admitting water slowly, the pressure gradually rose to a maximum and then fell back to a stationary point several per-cent lower than the maximum. This drop in pressure marked the rise of the valve. On account of the small quantity of water flowing, this rise was very slight. The following table shows results of the test.

Table 1.

Valve	Head to raise Valve Ft.	Wt. of Valve + Spring Tension Lbs.	Lifting Force-Lbs. Computed on Bottom Area	Top Area	Percent of cont. Area Effect.
A_1	1.71	5.35	4.53	7.50	72.5
A_2	0.92	3.94	2.43	4.02	5.0
A_3	1.42	4.30	3.75	6.20	77.5

These figures show that all the contact area of the seat is not effective, that is, it does not hold water.

2. Motor Rating

It was thought at first that rating curves for the motor and shafting might be established which could be used throughout the tests for determining the power put into the pump at different speeds. With this in view brake tests were made of the power delivered to the pump shaft for each of the five speeds, over a range of possible horse powers. It was felt however that full confidence should not be placed in these ratings without checking them up at frequent intervals. It was found in various attempts to check them that they were unreliable for good results on account of voltage fluctuations and other factors such as lubrication of bearings and tightness of belts. Thereafter

the brake data was taken immediately following the regular run, before conditions could change appreciably. The input to the pump has been taken from the rating curves for the purpose of comparing it with that obtained from regular brake data. The percent error in the rating curve has been computed and plotted in Plate 19 . The results show the futility of attempting to establish a series of rating curves for the motor and shafting which will apply within reasonable limits of error for tests over quite a period of time and when unavoidable changes in belt tension and shaft friction are sure to occur.

Difficulties

Some entirely new apparatus had to be designed for these experiments, and it was expected that defective design and construction would appear when the new parts were put into use. For the benefit of those who may wish to carry these experiments further, we shall discuss briefly the difficulties in the order in which they were encountered.

The target holders were designed for glass windows, the joints to be sealed with sodium silicate or water glass. This arrangement immediately gave trouble from leakage and the breaking of the glass. A waterproof

red wax was next tried for sealing the joints but was unsatisfactory. Finally, it was found that mica or celluloid for the windows used with paper gaskets was very satisfactory. It was found necessary just before the beginning of each run to loosen the face of the target holder in order to permit the escape of air bubbles which collect after the pump has stood idle for some time. These bubbles seriously diffused the light when left in.

The photographic machine was tried out in the mechanician's shop and worked very well. "Eastman's Velvet Bromide Paper" was found to give good results and it was used in all the tests. But after the targets were placed on the pump and had contained water for a considerable period, the records obtained were much too dim. The water had so stained the mica windows that the intensity of the arc had to be greatly increased. This was done by permitting more current to flow. In order to make the carbons burn evenly and so prevent the arc from becoming obscured, the carbons were set in line with each other instead of at an angle as was first done.

The motor for moving the paper in the machine was provided with an adjustable automatic centrifugal

governor. But neither this device nor the amount of current supplied would ever give satisfactory speed regulation. It was found however that the speed could be fairly well controlled by hand, and this practice was followed. It is now thought that the paper drum should be driven from the jack-shaft through flexible shafting or universal joints. This would give it a speed proportional to that of the pump, and hence all valve cards would be of equal length.

To operate the electromagnets properly about one ampere was required. This caused destructive sparking at the relay points. To prevent this, condensers were connected across the points. Probably on account of residual magnetism and leakage currents the armatures of some of the electromagnets were not released when the circuit was broken. This was avoided by placing a thickness of paper between magnets and armature, so as to prevent actual contact.

Air in the pump and its effect has already been mentioned. The amount entering was probably quite small, and it seemed that the best way to avoid this trouble was to permit all valves to operate and thus remove any pockets where it might collect. It should be stated here that no trouble of this kind was exper-

ienced with the short-circuit runs because the suction was under a low pressure.

CALCULATION OF DATA

Switchboard or Electrical Horse Power

The means of the ~~ten~~ readings of the ammeter and voltmeter were corrected by referring to the calibration tests. Then the product of volts and amperes was divided by 746 to give the horse power input of the motor, or $\frac{I \times E}{746} = E.H.P.$

Brake Horse Power

This quantity was calculated by using the mean of the two or more speed observations and the load on the scales in the formula $\frac{2 \times 3.14 \times 1 \times R.P.M. \times L}{33,000} =$

$$\frac{2 \times 3.142 \times 3.49 \times R.P.M. \times L}{33000} = .000666 \times R.P.M. \times L =$$

B. H. P. where L = brake load.

Efficiency of Motor and Shafting

The loss in motor, shafting and belting is evidently the difference in the input of the motor and the brake horse power or the input to the pump, and the efficiency of that part of the apparatus is the ratio $\frac{B.H.P.}{E.H.P.}$

Indicated Horse Power and Valve Loss

From the indicator cards were calculated valve loss, indicated horse power, and the power output of the pump. The cards were gone over with a planimeter for

area, and the mean height was obtained by dividing the area by the length. This mean height times the spring factor plus a calibration correction gave the mean pressure during the stroke. The pressure for the discharge stroke in the plunger chamber minus the pressure for the same stroke in the discharge chamber gave the pressure loss through the discharge valve.* In the same way the loss through the suction valves was obtained. The horse power due to this loss was the sum of the horse powerfor the suction and discharge decks. The power loss for one deck of one chamber is

$$\frac{p \times R.P.M. \times 3.142 \times d^2 \times 10}{33000 \times 4 \times 12} = .000714 \times p \times R.P.M.$$

= I. H. P., where p = algebraic difference of suction and discharge strokes.

It was found in the runs under pressure that the loss in pressure through the valves was very inconsistent and so it was decided that data from these runs was unreliable for estimating this loss chiefly on account of the relative strengths of the springs. The loss of head through the valves is such a small part of the total head, being about one pound per square inch at the fast speed, that it is practically impossible to scale this small quantity from the indicator card. Since our experiments show no effect on valve action

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*Corrected for difference in elevation of indicators.

due to change of pressure, it was thought proper to assume that valve loss was independent of pressure, and so calculations have been made from data obtained in the short-circuit runs.

Output Horse Power

The output of the pump was calculated from indicator cards taken on the discharge pipe, and the weir readings. To the mean pressure on the discharge card is added the suction head measured on the mercury column and a quantity for the difference in elevation of the vacuum gage and indicator. If q = discharge in second feet,

$$\frac{qh}{550} = \frac{q \times p \times 62.5 \times 2.304}{550} = .262 \times q \times p = \text{H. P. output.}$$

Pump Efficiency

The total pump efficiency is plainly the ratio:

$\frac{\text{Output H. P.}}{\text{Input H. P.}}$ The mechanical efficiency is the ratio:

$\frac{\text{Output H. P.}}{\text{Ind. H. P.}}$ The indicated horse power minus the output equals the total hydraulic loss in the pump, and this quantity minus the valve loss gives the loss in other passages of the pump. The hydraulic efficiency then is the ratio $\frac{\text{H. P. Output}}{\text{Ind. H. P.}}$

Photographic Cards.

The sensitised paper used in the photographic apparatus described before was in ten-yard lengths, having a width of 2 in. When developed the valve diagram and the cross head and time coordinations were in black upon the white strip. The valve diagram was a continuous line, while the cross head and time marks were merely short dashes above and below the valve diagram. Any movement of the valve was shown by the vertical component, while the passing of the paper from one drum to the other generated the horizontal component of the diagram. Three or four diagrams for one valve were obtained at one exposure.

The paper was not drawn *by* the target at an absolutely uniform speed, which would have been better. The small motor was not able to take care of the varying load put upon it, and different valve diagrams were obtained on different bases, thus making any two more complicated to compare.

For successive diagrams on the same valve during one exposure the spacing of the time marks was fairly uniform, thus indicating practically constant speed of the paper. For any particular stroke, however, the speed of the paper could be considered constant, as

the time and space passed over were small. It follows then that the valve lift diagrams are on a time instead of the usual stroke base.

Due to the construction of the machine, the cross head shutter was not directly over the valve target, but was a fixed distance from it. To study the cards the cross head marks had to be offset to the right in order to truly coordinate the valve motion with regard to the dead-centers. The cards were generated from right to left.

For each run the maximum valve lift of each valve was determined from the cards and also from a scale measurement taken at the target holder. The average lifts during the stroke, on a time basis, were calculated as follows. The area bounded by the valve diagram and the zero line was determined by a planimeter. The average lift was then computed by dividing this area by the distance between the dead-center marks on the cards. The correction for magnification was applied in both maximum and average lift data.

When once the dead-centers had been properly placed on the card, measurements were taken to see how nearly the valves opened and closed at the dead centers.

The time mark was not used in the calculations as the

speed of the pump during a run could be determined from the revolution counter.

In the study of instantaneous lift the stroke was divided into equal parts and by means of dividers the curve in question was reproduced on some other base. The relation of the diagram on a stroke and time basis is shown on Plate 15. The displacement or velocity curve of the plunger, the connecting rod and crank ratio being 6:1, has in some cases been plotted to show the relation between the lift and velocity of the plunger. For explanation of piston velocities see "Steam Engine Design" by J. H. Whitham.

RESULTS

Losses and Efficiencies

The various horse powers and efficiencies for pump and motor are shown in the accompanying tables and curves. It is seen that the efficiency of the motor and shafting varies with both speed and power. One run gave a point of 83.4 percent, 8 percent above any other value. We are unable after checking the calculations to account for such a high figure. Minor variations, however, are due to differences in voltage, condition of brushes and commutator, tightness of belts, and lubrication. The voltage particularly was variable, ranging between the extremes of 430 and 570 volts. The loss of power between the switchboard and pump is dependent upon two things; first, on the efficiency of the motor; and second, on the frictional losses in the belting and shafting. The efficiency curve of the motor is plotted on Plate 18.

Note:- Data for this curve was taken from a thesis in the University of Wisconsin Library entitled "Efficiency Test on a Deming Triplex Power Driven Pump" by R. W. Muckelston and K. Slidell, 1909.

It is seen from the curve that the efficiency does not vary much for loads from 50 to 100 percent of the motor capacity (25 H. P.), but below 50 percent capacity the efficiency will change appreciably for relatively

HORSE POWERS AND EFFICIENCIES

Run		R.P.M.	Dischg. Press. Lbs.	H. P. Input to Motor	H. P. Input to Pump	Ind. H. P.	Pump Frict. H. P.
15	A	40.0	13.5	9.42	5.10	2.01	2.99
"	B	44.3	"	9.75	5.75	2.30	3.45
"	C	51.6	"	10.5	6.30	2.71	3.59
"	D	57.4	"	10.7	6.10	3.07	3.03
"	E	67.5	"	12.8	7.40	3.55	3.85
30	A	39.0	28.2	12.5	8.60	3.51	5.09
"	B	44.3	"	13.3	9.40	4.07	5.33
"	C	50.9	"	14.1	10.2	4.68	5.52
"	D	59.5	"	15.5	11.0	5.33	5.67
"	E	66.8	"	17.1	11.6	6.29	8.51
45	A	36.3	42.5	12.7	9.2	4.59	4.61
"	B	41.3	"	13.8	10.4	5.26	5.14
"	C	49.1	"	15.8	11.7	6.34	5.36
"	D	59.7	"	19.0	14.1	7.98	6.22
"	E	65.0	"	21.4	15.7	8.66	7.04
60	A	36.0	59.0	13.6	9.80	6.12	3.68
"	B	41.4	"	15.0	11.2	6.86	5.34
"	C	48.4	"	16.7	12.5	8.28	4.22
"	D	58.7	57.5	20.5	15.0	9.83	5.17
"	E	64.7	"	21.9	15.8	10.8	4.96
75	A	36.0	74.5	17.4	13.1	7.63	5.47
"	B	40.7	"	18.0	15.0	8.48	6.52
"	C	47.9	"	20.0	14.7	10.4	4.22
"	D	55.6	"	24.3	17.9	12.3	5.56
"	E	61.4	"	26.3	19.1	13.5	5.61

HORSE POWERS AND EFFICIENCIES

Run	H. P. Output of Pump	Motor & Shaft Eff.	Mech. Eff. of Pump	Hyd. Eff. of Pump	Total Eff. of Pump	Total Eff OverAll
15 A	1.82	54.2	39.4	90.5	35.7	19.3
" B	2.06	59.0	40.0	89.5	35.8	21.2
" C	2.40	60.0	43.0	88.6	38.1	22.8
" D	2.66	57.0	50.3	86.8	43.6	24.8
" E	3.17	57.8	48.0	89.5	43.0	24.8
30 A	3.32	68.8	40.8	94.6	38.6	26.5
" B	3.73	70.7	43.2	91.6	39.7	28.0
" C	4.33	72.4	45.8	92.5	42.4	30.7
" D	4.97	71.0	48.5	93.2	45.1	32.0
" E	5.69	67.9	54.1	90.5	49.0	33.2
45 A	4.50	72.4	49.9	98.0	49.0	35.4
" B	5.23	75.4	50.6	99.4	50.2	37.9
" C	6.05	74.1	54.1	95.5	51.7	38.3
" D	7.57	74.2	56.5	95.0	53.7	39.8
" E	8.29	73.5	55.2	95.5	52.7	38.7
60 A	5.70	72.0	62.5	93.1	58.2	41.8
" B	6.38	74.7	60.4	93.0	57.0	42.5
" C	7.60	74.9	66.2	91.8	60.7	45.5
" D	8.83	73.2	65.5	88.8	58.8	43.0
" E	9.96	72.2	68.7	91.8	63.0	45.5
75 A	6.79	75.4	58.2	89.0	51.7	39.0
" B	7.[illegible]1	83.4	56.5	93.4	52.7	44.0
" C	9.34	73.5	70.5	90.0	63.5	46.7
" D	11.1	73.7	69.0	90.0	62.0	45.7
" E	12.1	72.6	70.5	89.9	63.4	46.0

small changes in load. The losses in the belting and shafting increase gradually with increase in load but probably at a very nearly uniform rate. The voltage fluctuations mentioned above probably affect the efficiency of the motor to an appreciable extent below 50 percent capacity (10 or 12 H. P.)

On Plate 19 we have shown the percent errors which would have been made by taking the set of rating curves made at the first of the experiments as correct, and using them to determine the horse power delivered to the jack-shaft of the pump.

From a study of this plate we see that for loadings of 50 percent or greater an error of 5 or 6 percent is quite likely to occur. For motor loads under 50 percent of the rated capacity the errors would be much greater. In this latter case the larger error of reading smaller quantities enters as well as the errors due to the fact that the motor is working on a steep part of its efficiency curve where small changes in load or other disturbing influences make large changes in efficiency.

The mechanical efficiency depends largely upon packing friction. This was found to be a great variable, and uncontrollable to a large extent. The packing glands had to be adjusted occasionally for changes of

pressure, but this was nearly always done between runs of different pressures. As an instance of varying packing friction runs A75 and B75 give efficiencies 12 percent below the other runs at 75 pounds pressure. In general, mechanical efficiency increases with both speed and pressure.

The hydraulic efficiency of the pump is the only efficiency which does not increase consistently with either pressure or speed, perhaps on account of the small difference between the indicated horse power and the output of the pump, and because of the limited accuracy of indicator records. The runs A45 to E45 give very high hydraulic efficiencies. These were check runs and gave somewhat better figures than the original runs. But although the indicators were calibrated for this pressure the figures were apparently high as compared to all other hydraulic efficiencies.

The total efficiency of the pump is the product of the mechanical and hydraulic efficiencies. It tends to increase with both speed and pressure. It may seem that 63.5 percent, the highest value obtained, is rather low for this type of pump. It should be remembered here that the pump was designed for 7-inch plungers whereas it was equipped with 6-inch ones. The friction force between plunger and packing varies

as the diameter, while the capacity of the pump varies as the square of the diameter. Therefore, since plunger friction absorbes a very great part of the power lost in a pump, if the 7-inch plungers had been used, a higher efficiency would have been obtained.

The efficiency over all is the ratio of pump output to motor input, or the product of the hydraulic, mechanical, and motor and shaft efficiencies. It also increases with speed and pressure.

The valve loss, considered independent of pumping head, is calculated in horse power for the five speeds used. The percentage this loss is of the hydraulic losses and of the horse power input of the pump is also calculated and shown in the following table. It is seen that the valve loss is a very small horse power, and a small percentage of the total energy input of the pump. As would be expected, however, the valve loss is a very considerable percent of the hydraulic losses. The percentage in each case seems to increase with speed and pressure. Two percentages of the hydraulic losses run above 100 percent. These are to be taken as bad points. Mention has already been made of the uncertainty of hydraulic losses calculated with 45 lbs. pressure.

Speed R.P.M.	Valve Loss H.P.	Valve Loss in Percent of Hydraulic Losses					Valve Loss in Percent of Pump Input H. P.				
		P=75	P=60	P=45	P=30	P=15	P=75	P=60	P=45	P=30	P=15
36 - 40	.132	15.7	31.4	147.0	69.5	69.5	1.01	1.35	1.44	1.54	2.59
40.7-44.3	.139	40.0	29.0	465.0	41.0	58.0	0.93	1.24	1.34	1.48	2.42
47.9-51.6	.169	16.3	24.8	57.3	48.4	54.5	1.15	1.35	1.44	1.66	2.68
55.6-59.7	.216	17.4	21.6	52.7	60.0	52.7	1.21	1.44	1.53	1.96	3.54
61.4-67.5	.281	20.2	32.0	76.0	47.0	74.0	1.47	1.78	1.79	2.42	3.80

Slip

The term "slip" is ordinarily taken to be the ratio of the difference between displacement and discharge to the displacement. It thus includes valve slip and all leakage. The results are given in the table below.

Pressure Lbs. per Sq.In.	Speed R.P.M.	Slip Percent	Pressure Lbs. per Sq.In.	Speed R.P.M.	Slip Percent
75	36.0	3.19	30	39.0	1.72
	40.7	2.10		44.3	1.94
	47.9	2.01		50.9	1.69
	53.6	1.72		59.5	1.95
	61.4	2.80		66.8	1.72
60	36.0	3.19	15	40.0	2.63
	41.4	3.00		44.3	1.51
	48.4	2.97		51.6	1.48
	58.7	2.94		57.4	1.50
	64.7	1.49		67.5	1.42
45	36.3	3.21			
	41.3	2.12			
	49.1	2.55			
	59.7	2.53			
	65.0	2.47			

Valve Diagrams

On Plates *12, 13 and 14* are tracings of some valve diagrams from the photographic records. Representative diagrams both in the short-circuit and regular runs have been taken to illustrate certain features in the operation of the pump. In all these typical diagrams the dead center marks have been put in. Due to

the fact that it was inconvenient to construct the apparatus in such a way that the cross head and time shutters would be directly above and below the target shutter, respectively, it was necessary to offset these marks a fixed distance before studying the diagrams. The vertical lines thus mark the dead centers, and the lift records are generated from right to left. No cross head marks are shown for the valves in deck C because the contact was fixed only on the A and B side of the pump and the variable speed of the motor made it inconvenient to get them for the C deck. The dotted lines on the C diagram, however, mark very closely the dead centers. The time marks are absent because of relay trouble and besides the speed of the pump was obtained from the revolution counter.

In the short-circuit runs the pressure discharged against was low, while the suction was under an actual pressure from the small standpipe in the short-circuit. Taking up the low speed on the sheet of tracings, the valves in decks A and B are seen to rise slightly after the beginning of the stroke while the valve in deck C is seen to rise just a little before. Tho the suction was under pressure, the valves in A and B failed to rise just at the point of reversal for reasons

given under the regular runs. The valve in deck C had a weak spring and the retardation of the plunger on the suction stroke together with the positive suction pressure was sufficient to cause the valve to rise before the actual discharge stroke had begun. With the high speeds these effects were more pronounced, especially in the valve C_3.

Taking up the closing of the same valves under the same conditions it is seen that at the low speeds A and B valves close just about at the end of the discharge stroke, while C does not close promptly. This is explained as follows:

The pump was double acting and when the discharge stroke on one end was just completed the suction stroke on the other end of the plunger was just completed. The two chambers had a common suction pipe. Thus the retardation at the end of the suction stroke on one side (even tho its own discharge valves were opened prematurely) had the effect of causing a rise of pressure in the common suction main sufficient to keep the valve on the other side of the plunger from closing immediately. The inertia effects of the water in the clearance in the discharge chamber had also some slight tendency to keep the valve open. As the speed increases

this is seen to increase due to the greater magnitude of the retardations and the increased lift of the valve. When discharge valves act this way overdischarge takes place, i. e. the actual quantity of water discharged by the pump is greater than the actual displacement of the plunger would indicate. This phenomenon only occurs in practice of course with comparatively high speeds, very low discharge pressure and positive suction pressure.

The discussion of the cards in the regular runs will now be taken up. Perhaps the first thing a person would notice is the difference in the diagrams of valves in the same deck as regards form, maximum height, and time of rising and closing. All these differences are caused by differences in weight, initial spring tension and strengths of springs. One would expect, at first thought, that at least one valve in each deck would leave its seat at the beginning of the discharge stroke. Such is not the case according to the photographic cards. From them we are led to believe that 0 - 6% of the stroke on a time base, or about 0 - 4% of the actual stroke, may be completed before any valve rises in a deck. No very consistent increase in the lag is noticeable with an increase of pressure or a decrease of speed, although this lag varies from zero

up to the maximum, as stated. Due to differe t spring tensions, etc., the three valves do not rise together. The valve having the stiffest spring, highest initial tension and weight will always be the last to rise and the first to close if working freely. It was found in some cases that one valve remained on its seat until 20% of the stroke was completed, the other two valves being able to take care of the discharge at the beginning and end of the stroke where the instantaneous discharge was small. Of course it would be possible to give a valve such an initial loading, (with more than one valve acting), that it would fail to rise at all.

The reasons why the cards do not show at least one valve in each deck rising promptly on time are as follows:

1- Air may be entrapped and prevent an immediate rise of pressure. The indicator cards showed this lag of pressure at times. (See indicator cards).

2- Some suction valve in the chamber may not seat immediately from some cause and thereby allow a small slip thru the suction valve at the beginning of the discharge stroke.

3- Slight lost motion of apparatus, or rises too small to be observed on cards.

4- Errors in measuring cards.

The time of valve closing seems to be more consistent than the opening of the same. In all the cards taken in the regular runs, one valve in each deck seemed to close just at the reversal of stroke, the other two valves having seated before the end of the stroke. No valve in the regular runs was found to close by an appreciable amount after the end of the stroke. This fact is borne out by the small amount of slip of the pump. (See Theory for angle of lag.)

With a single discharge valve equation (32), page 22, h_{max} = (approx) $\frac{Frw}{ml\sqrt{2gb}}$, gives the maximum valve lift in terms of the angular velocity of the crank and the head "b" causing the discharge through the valve. Of the other factors entering the equation m is the only variable. From the curve, Plate 47, between m and h (taken from the article by H. Berg referred to above) it is evident m is essentially constant for the range of values which the maximum lift will have.

We may simplify the above equation then by putting

$$\frac{Fr}{ml\sqrt{2g}} = K = \text{a constant}$$

(35) Hence $h_{max} = K\frac{w}{\sqrt{b}}$

For simplicity let $h_{max} = H$

From equation (34) page 24

$$\sqrt{b} = \sqrt{\frac{W + (y_0 + H)C}{fY}} = \sqrt{\frac{W + y_0 C}{fY} + \frac{H\,C}{fY}}$$

Call $$\frac{W + y_0 C}{fY} = B = \text{a constant}$$

$$\frac{C}{fY} = B_1 = \text{a constant}$$

Then $$\sqrt{b} = \sqrt{B + B_1 H}$$

Substituting this value of b in the equation above and squaring

$$H^2 = \frac{K^2 w^2}{B + B_1 H}$$

$$(36)\quad BH^2 + B_1 H^3 = K^2 w^2$$

An equation between maximum lift and speed of rotation of the crank shaft.

The expression for this curve could probably be put in more simple form if desired by shifting the origin.

The above expression applies to a pump having a single discharge valve and should therefore apply to our short-circuited run with one valve working.

With only one discharge valve operating in each deck of the pump used for these tests, however, the action is modified. With the restricted valve opening used, the speeds obtainable from the motor were, with the exception of the lowest, great enough to force the

valve up to its limit, or against the safety stop.

Plate 40 , shows the curves between total maximum lift in deck A and the speed of the crank shaft, for the conditions of one, two, and three valves operating, when the pump is short-circuited.

The curve for one valve operating becomes a horisontal line as soon as the speed increases enough to lift the valve to the stop.

It will be noticed that the total maximum lift is not the same with a different number of valves working for any given speed. Two causes contribute to this result. First, changing the valve seat area relative to the piston area. Second, the spring loading of the valves is different.

From equation (1), page 4,

$$Fu = f_1 c_1 \text{ or } c_1 = \frac{F}{f_1} u$$

we see that reducing the area of the valve seat relative to the piston area increases the velocity of discharge through the valve seat. Increasing f_1 relative to F will have just the opposite effect. And from the equation at the bottom of page 15

$$\frac{P_1}{f_1 Y} = Z_1 \frac{c_1^2}{2g}$$

we see, assuming Z_1 practically a constant for the

condition of maximum valve lift, the force tending to open the valve varies as the square of the velocity through the valve seat.

The amount and nature of the loading which a valve has will also affect the maximum lift.

From equation (35), page 67

$$h_{max} = K \frac{w}{\sqrt{b}}$$

substituting for b its value in terms of valve loading from equation (21) page 18 we have

$$h_{max} = K \frac{w}{\sqrt{\frac{W + S}{fY}}}$$

If then the valve loading (W + S) is increased the maximum lift will be decreased, and if the valve loading is decreased the maximum valve lift will be increased.

Plate /6 page has been taken from H. Berg's article, referred to above, for the purpose of showing these two effects. All four diagrams were taken on a pump having a single discharge valve and running at 61 revolutions per minute. Diagrams 1 and 3, likewise 2 and 4, were taken with the same springs but with different plunger strokes. Changing the plunger stroke will change the plunger displacement, and therefore the velocity of discharge through the valve seat.

The maximum valve lift is seen to increase with the increased velocity through the valve seat.

Diagrams 1 and 2 or 3 and 4 were taken with different springs but with other conditions unchanged. These show the effect of valve loading on the maximum lift.

Referring to the maximum valve lift curves again we conclude the maximum valve lift increases according to some law with an increase in the crank speed; and that the nature of this law is modified by the ratio of the valve seat area to piston area, the loading of the valves, and the position of the safety stop. In the experimental pump the initial valve spring tensions were not the same and the increase of loads per unit compression of the springs were not identical. This results in complicating the law of action when several valves are present in one deck.

We had planned to make pressure runs with only one valve in each deck operating in order to compare results with one valve to those with several. An unexpected difficulty presented itself with this arrangement. The construction of the pump is such that with one or more valves blocked to their seats air pockets are formed in the plunger chamber which fill with air, when the pump is working on suction lift, and completely change conditions. The valve lift diagram on Plate 7

page was taken with this arrangement and shows the very late opening effect obtained when air is present in the plunger chamber. Had we been able to put the suction under pressure this could have been avoided. By making new pistons for the indicators this effect was reduced.

Plates 41, 42, and 43 show the relation between the sum of the three maximum valve lifts in the deck and the revolutions of the crank shaft for decks A, B, and C.

It will be noted that discharge pressure has no influence on the action of the valve so far as the maximum lift is concerned, and in fact any way that can be detected by comparing the valve diagrams.

The theory of valve action given is based on a constant discharge pressure, and since the experiments show that discharge pressure has no influence, it will apply to any discharge pressure.

The difference in the form of the curve for the three decks shows the effect of unequal valve loading on the action of the valves of the deck as a whole.

Plates 44, 45, and 46, are curves between the total mean or average opening in a deck and the crank speed. They follow the same general form as the corresponding maximum lift curves, and if the latter is known the nature of the average opening curve can probably be judged from it.

CONCLUSIONS

We conclude that the friction loss in the packing for the pump tested is the largest loss and the one subject to most variation. That adjustment of the packing glands may cause a change in this loss greater than all the other losses together. That the losses through the valves of the pump are the principal portion of the loss due to the passage of the water through the pump. That the method used heretofore of obtaining the power input to the pump is subject to errors as great as 5 or 6 percent.

We find that the action of the valves is not changed by varying the discharge pressure, that the maximum lift and mean opening increase with the speed of the pump, and that this law of increase may be altered by a change in the valve loading, or in the ratio of the valve seat area to the piston area.

We also found that the pressure necessary to raise a valve from its seat is not as large as would be calculated by taking the unbalanced area into consideration.

We found the valve loading to differ materially for the different valves in the pump used. It seems probable that good results with regard to low loss of head through the valve and quietness of action can only be obtained where due care is taken to have the valve springs as nearly

identical as possible when in position in the pump.

From the valve diagrams we find the discharge valve does not open immediately at the beginning of the discharge stroke but that it tends to seat just at the end of this stroke or very soon after.

We found that with the suction under a positive pressure and with the discharge under a low head the valves rise before the beginning of the discharge stroke and seat an appreciable time after the point of reversal. This effect is more pronounced at the higher speeds. In other words, under certain conditions of suction and discharge head and speed, a reciprocating pump of this character may pump more water than the displacement of the plungers would indicate.

With the suction pressure less than atmospheric the slip was found to be small, from 1 to 3 percent.

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Plate A

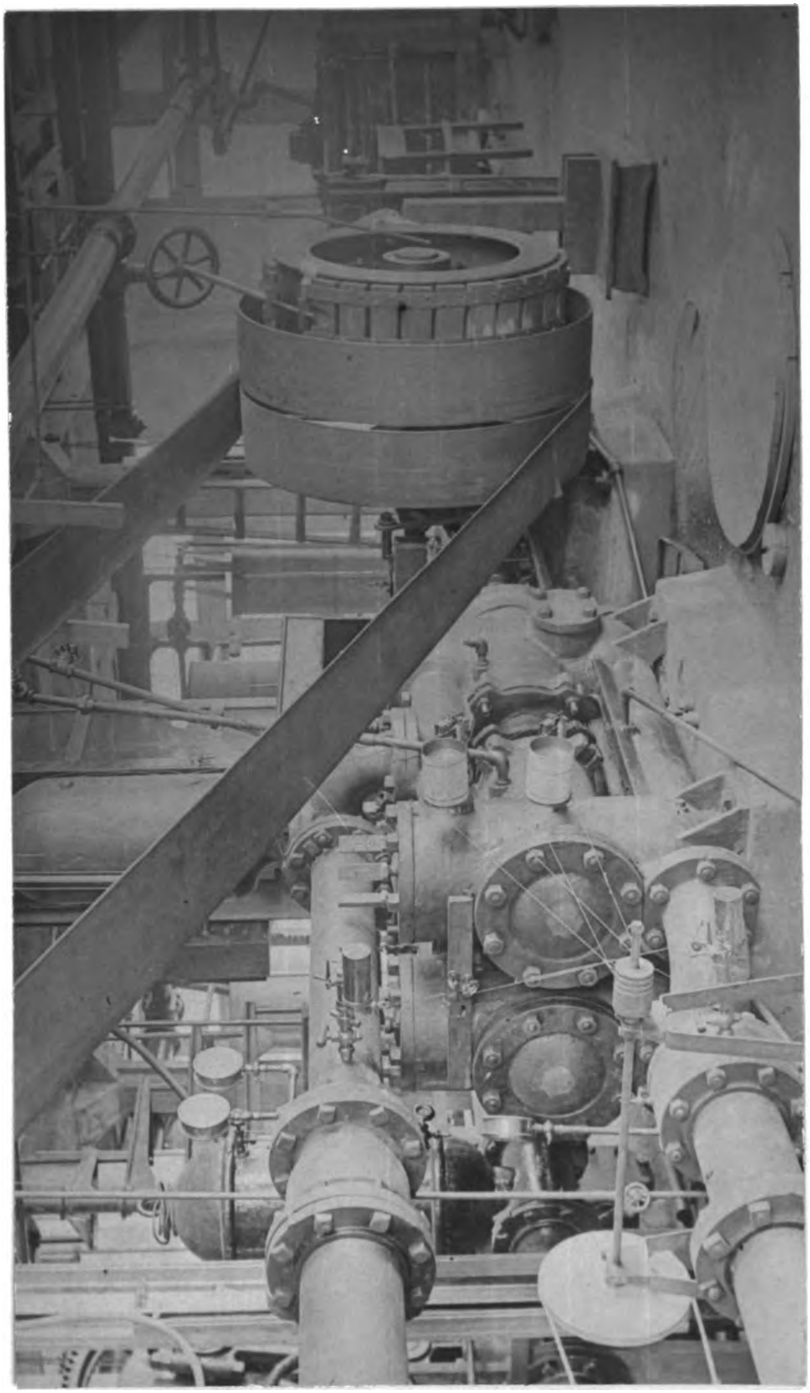
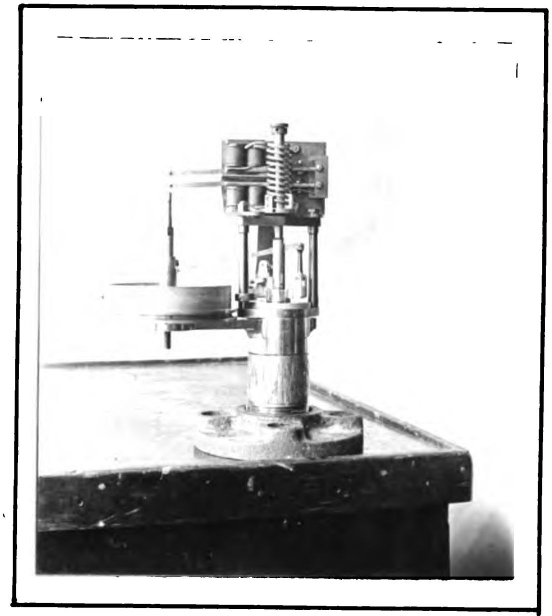
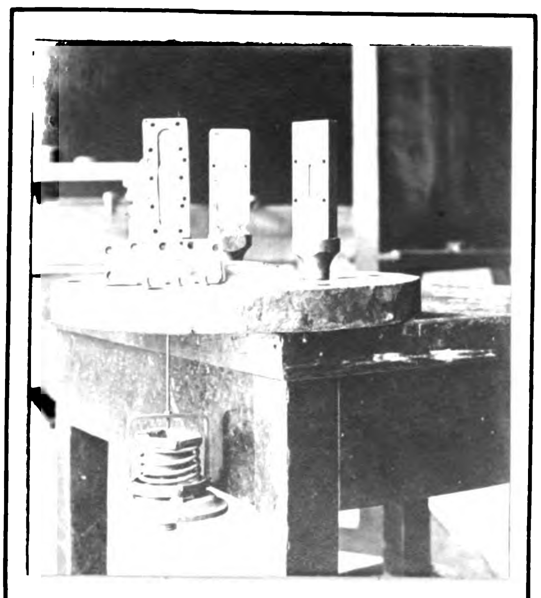


Plate B





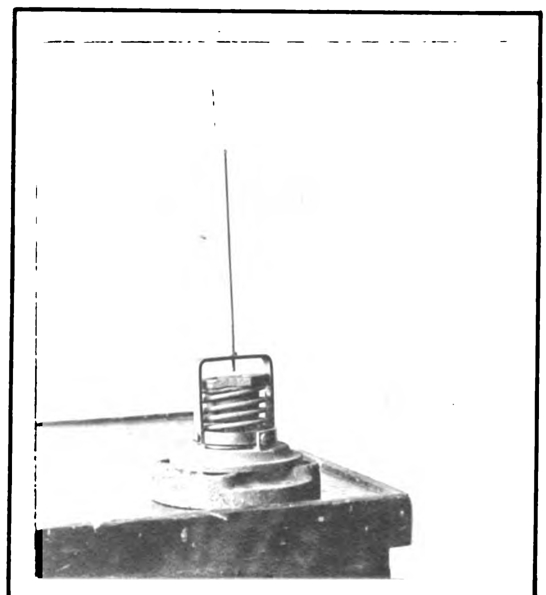


Plate C

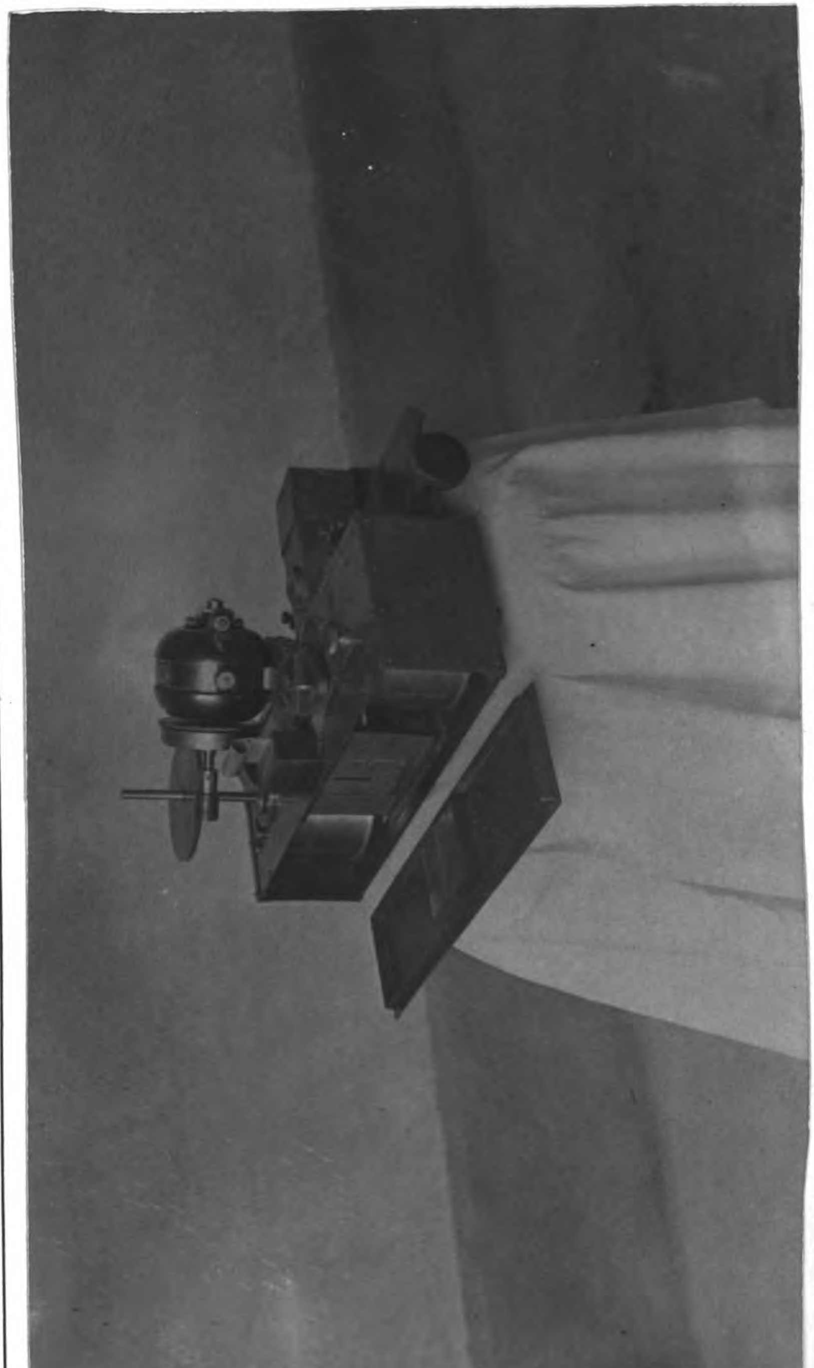
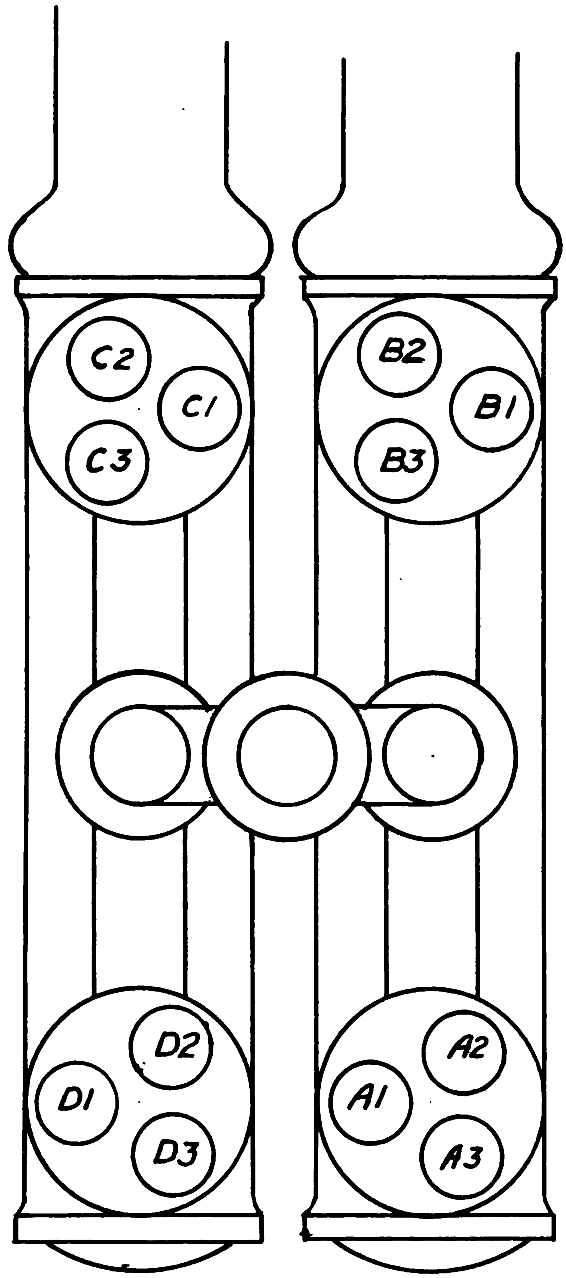
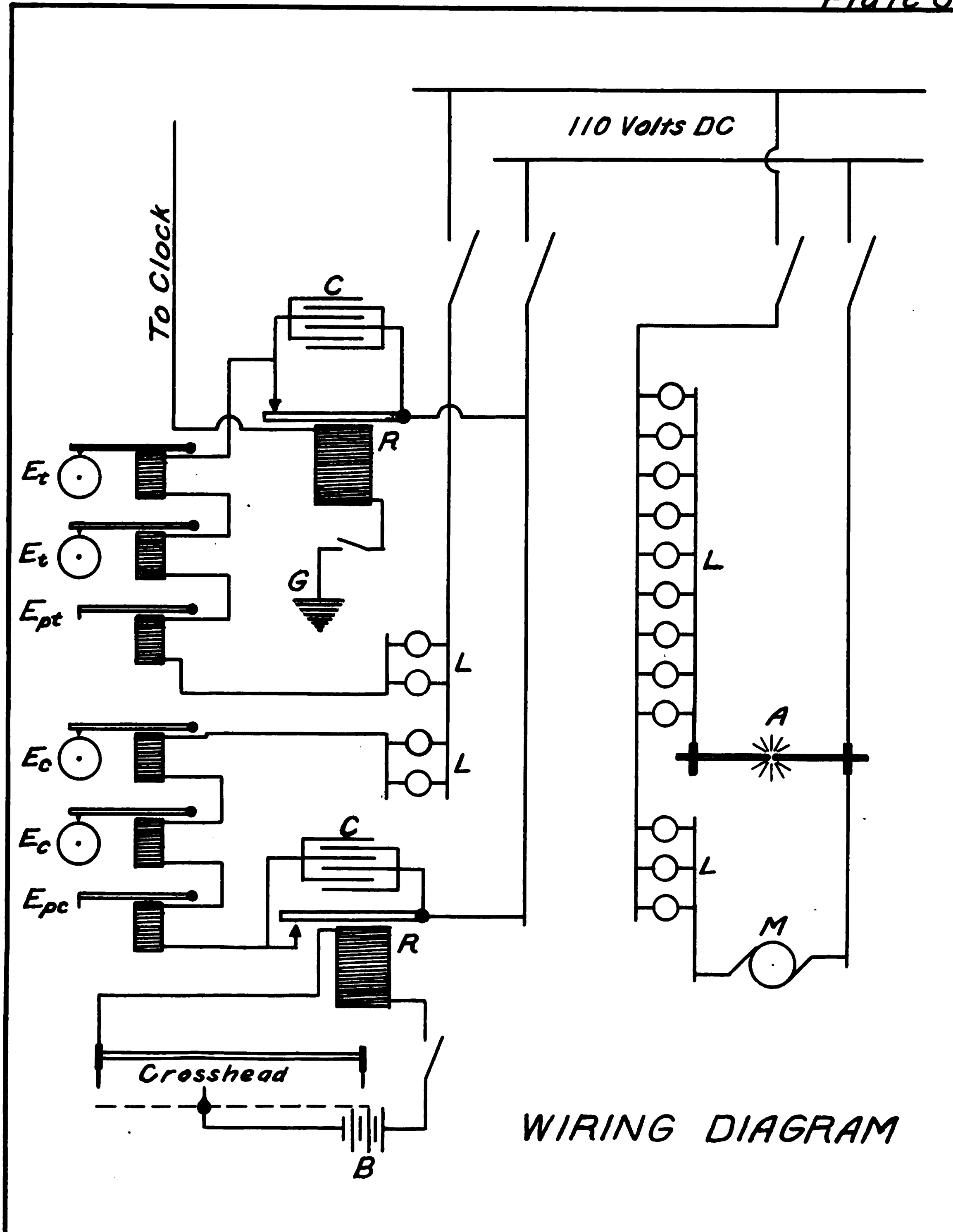


Plate I.



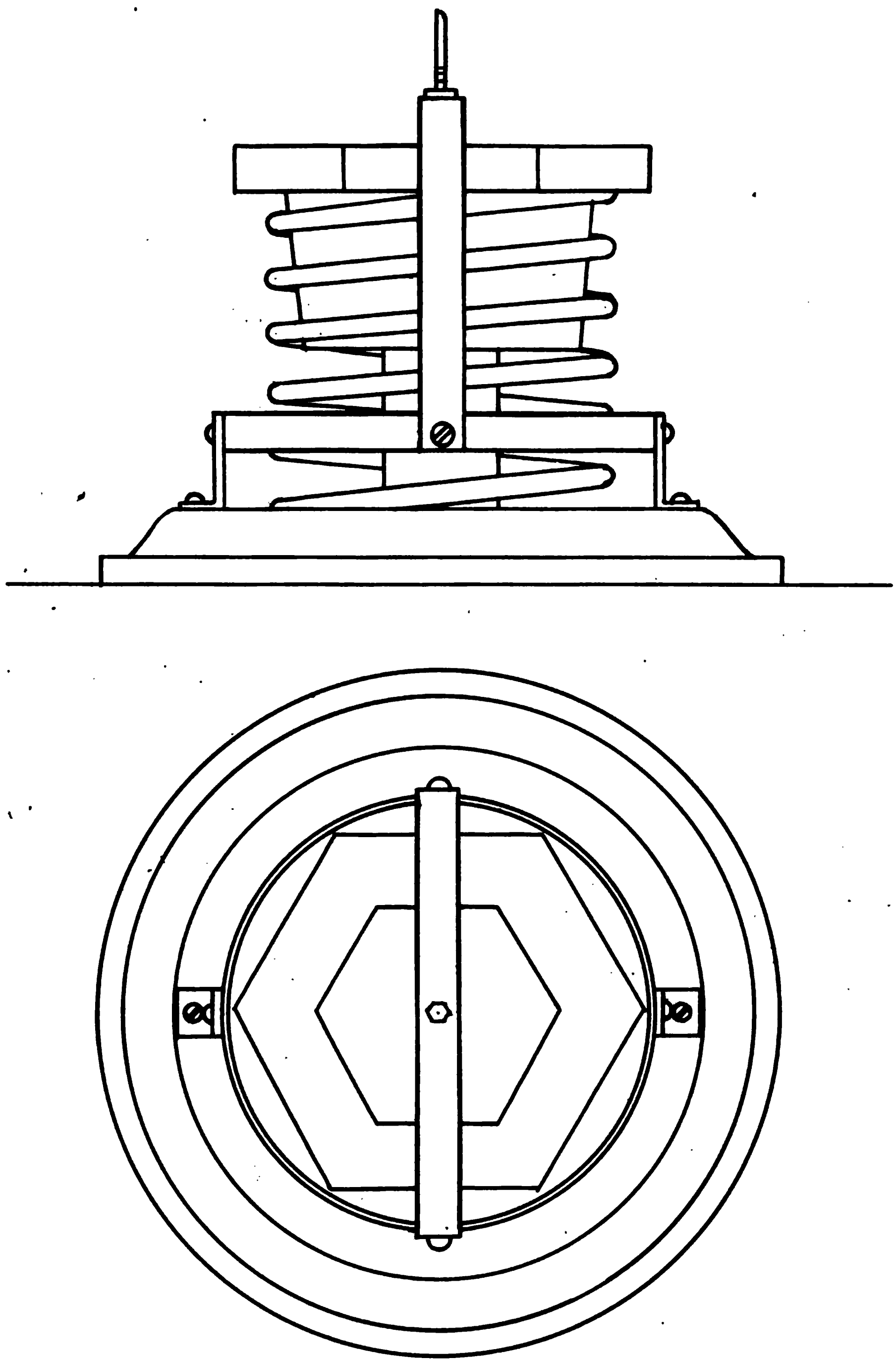
ARRANGEMENT OF VALVES

Plate 3

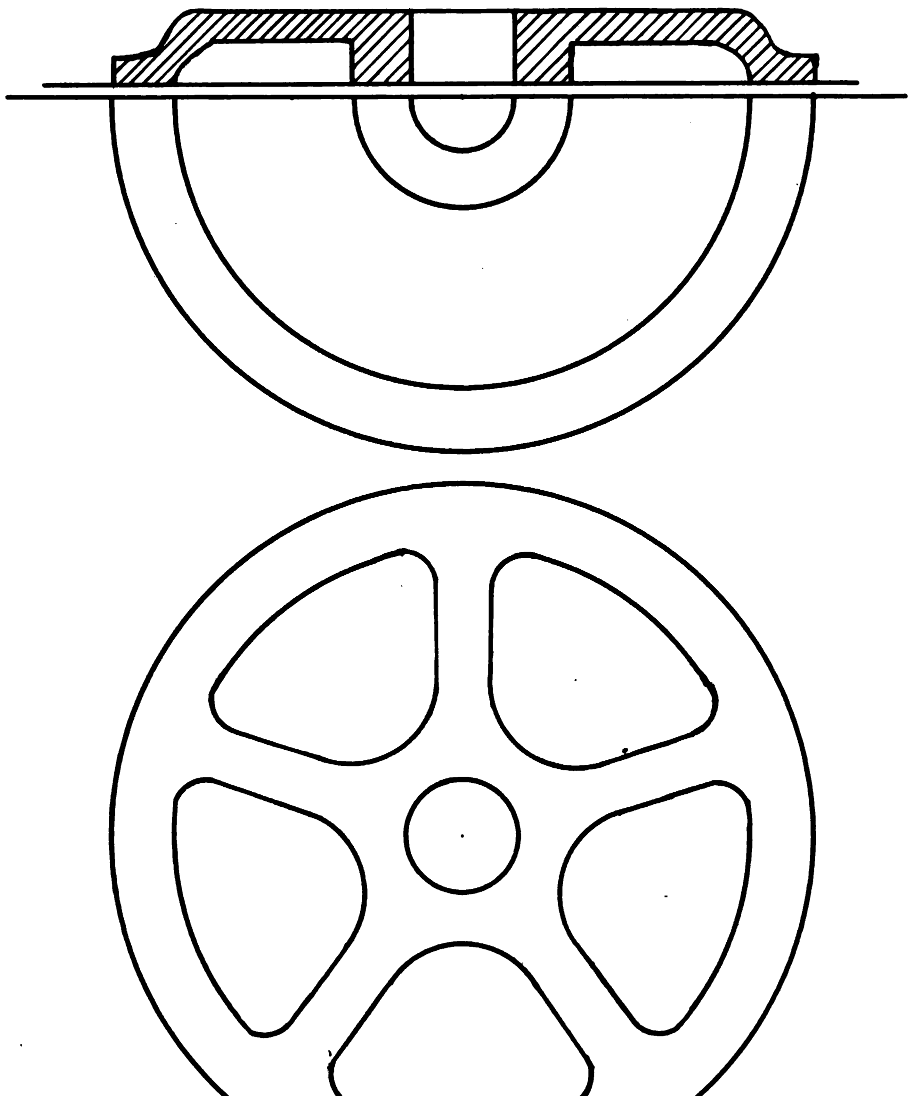


WIRING DIAGRAM

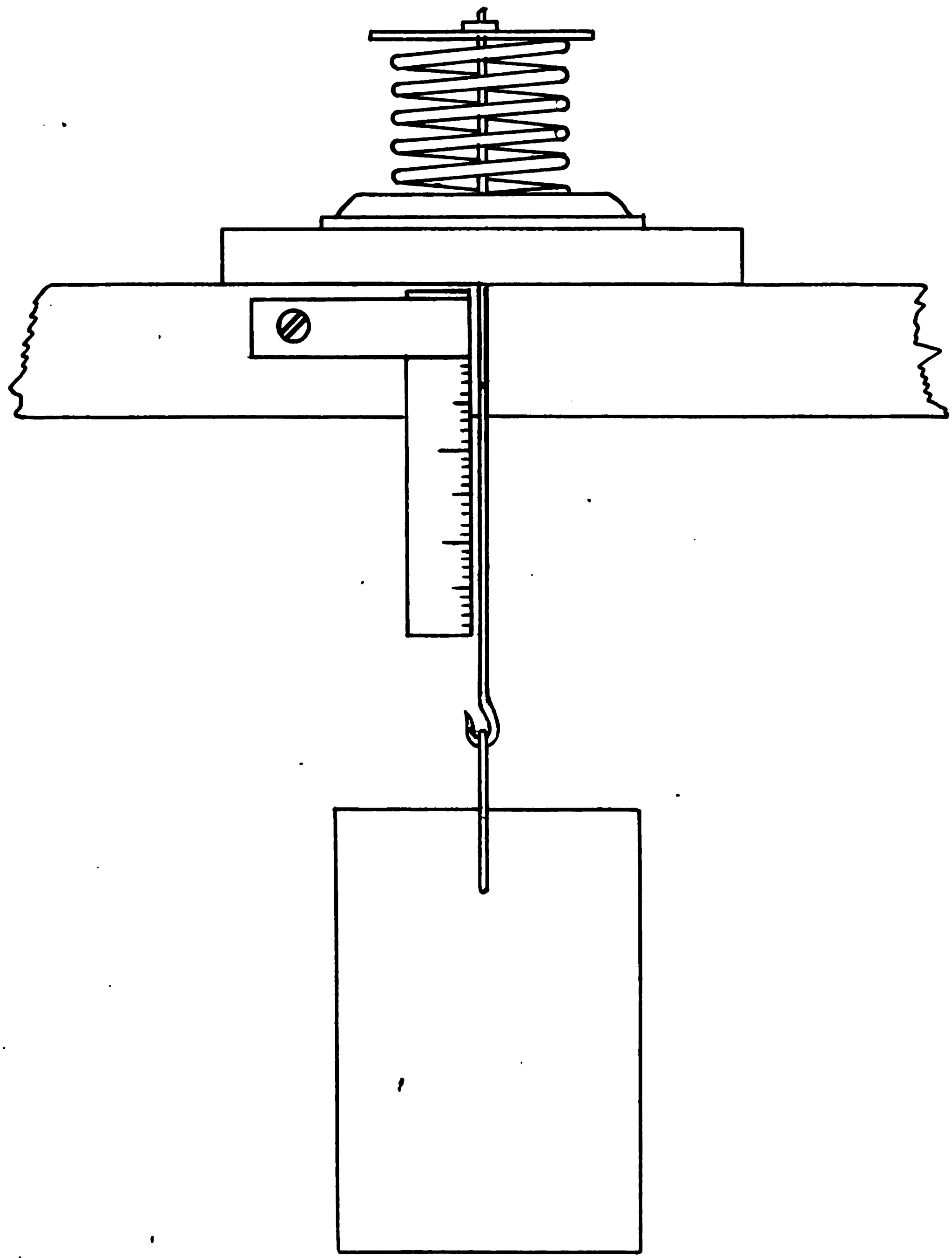
VALVE AND ATTACHMENT



VALVE AND VALVE SEAT

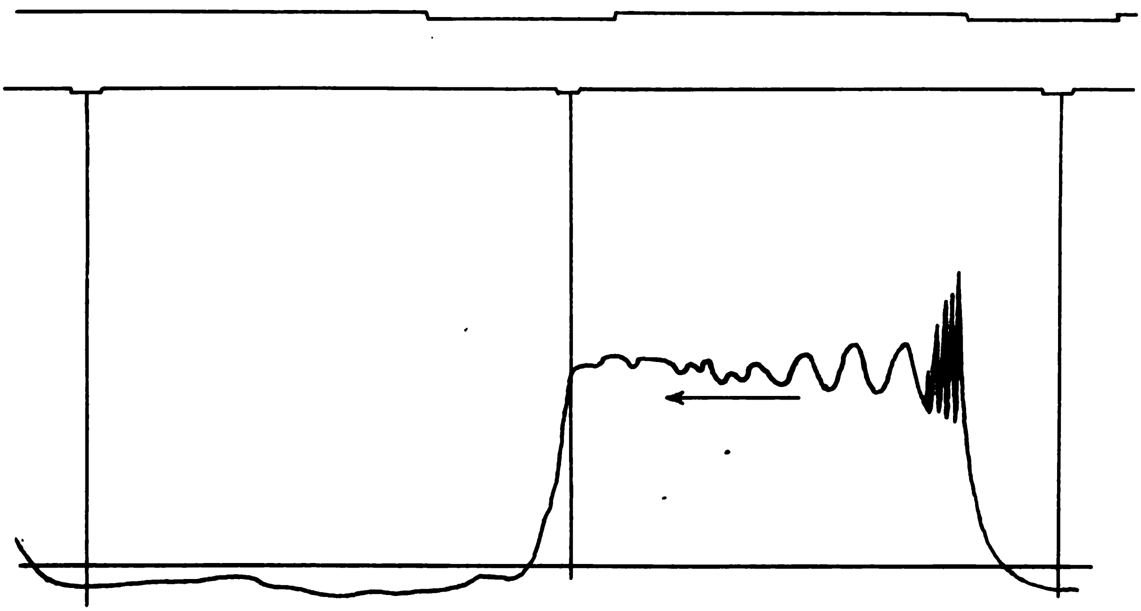


METHOD OF CALIBRATING VALVE SPRINGS

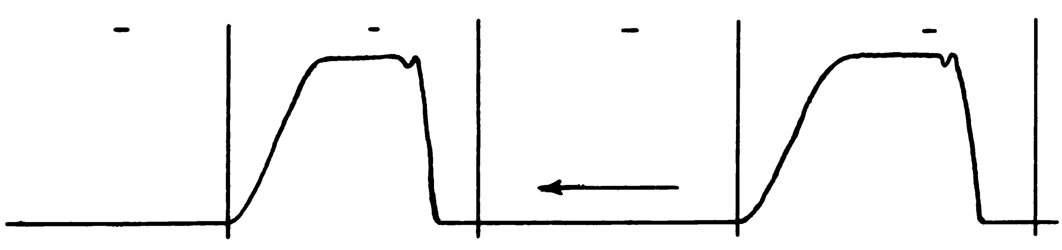


DIAGRAMS SHOWING EFFECT OF AIR IN PLUNGER CHAMBER

Indicator Card
60# Pressure



Valve Diagram
Speed B
One Valve Operating



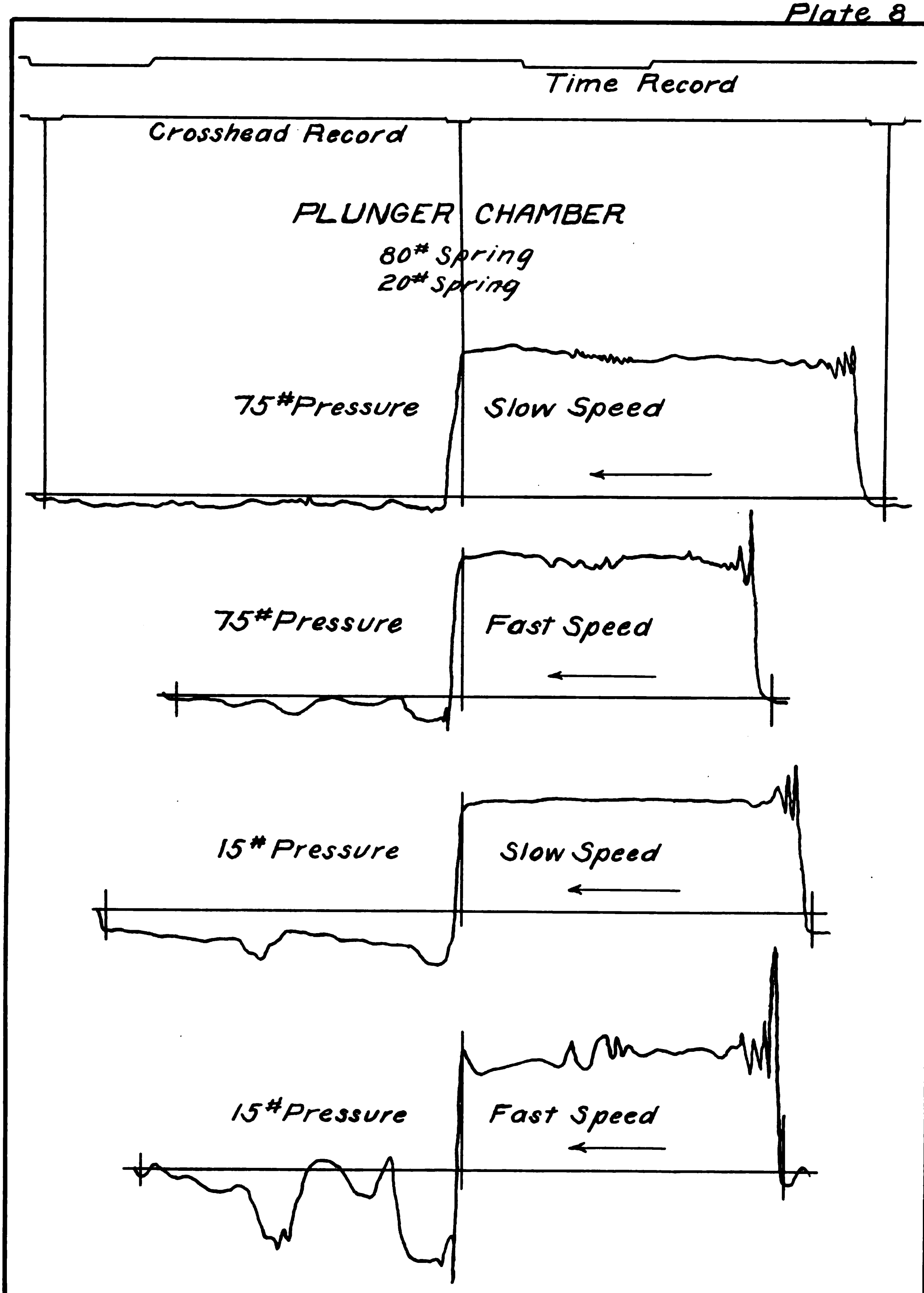
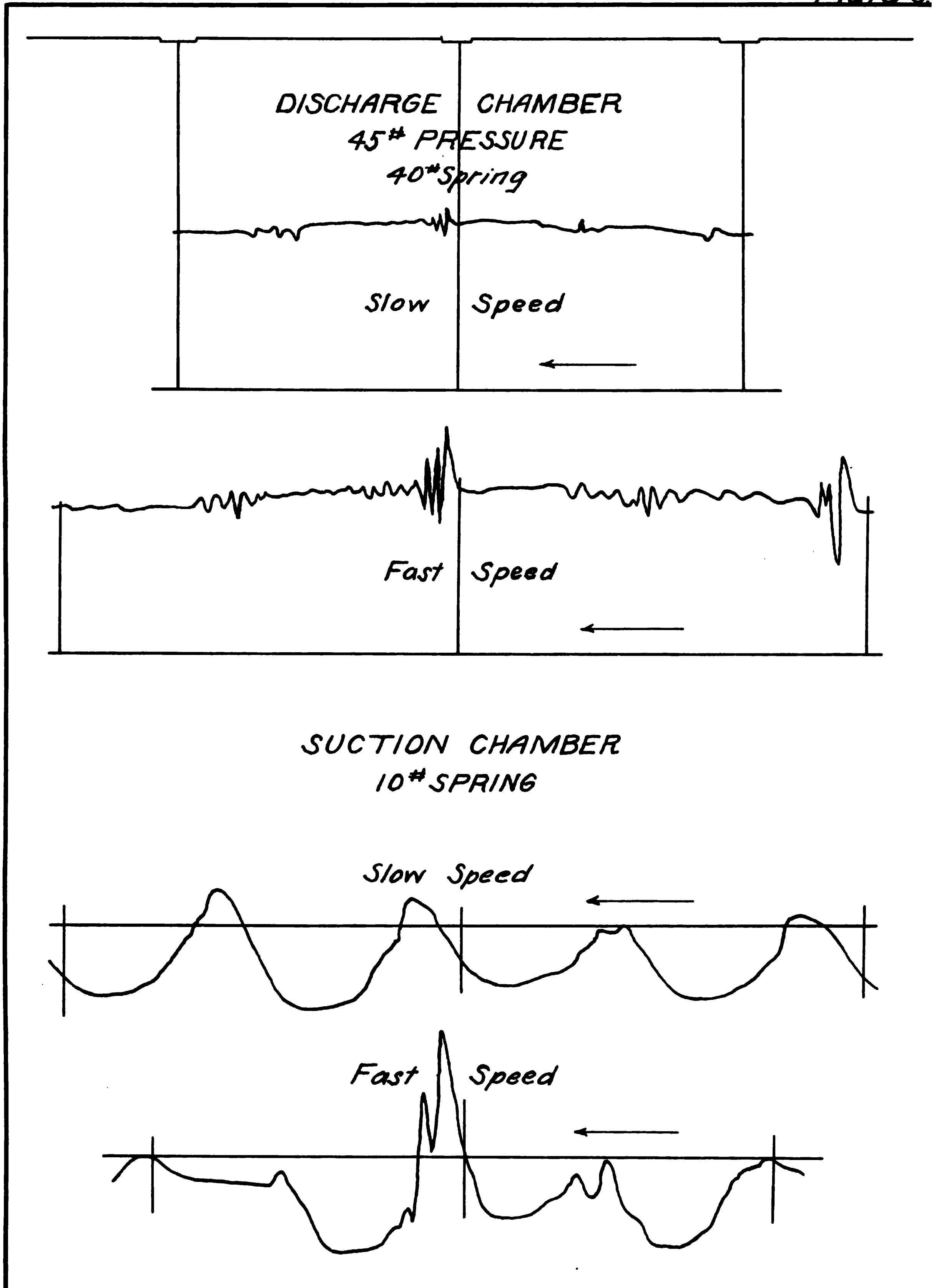
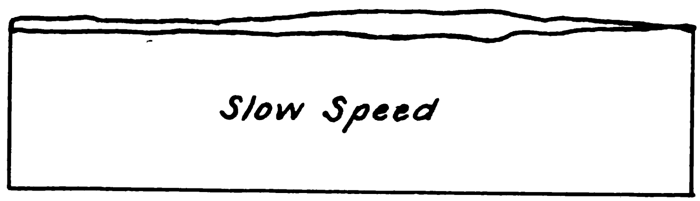
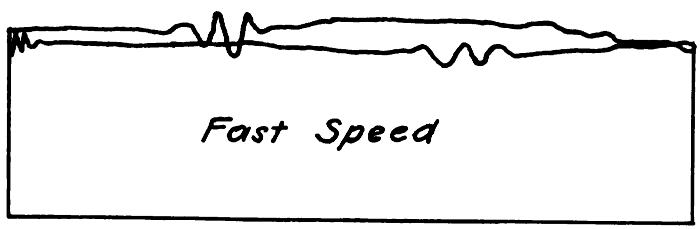

Time Record
Crosshead Record
PLUNGER CHAMBER
80# Spring
20# Spring
75# Pressure
Slow Speed
75# Pressure
Fast Speed
15# Pressure
Slow Speed
15# Pressure
Fast Speed

Plate 9.


DISCHARGE CHAMBER
45# PRESSURE
40# Spring
Slow Speed
Fast Speed
SUCTION CHAMBER
10# SPRING
Slow Speed
Fast Speed

DISCHARGE PIPE
75# PRESSURE
80# Spring

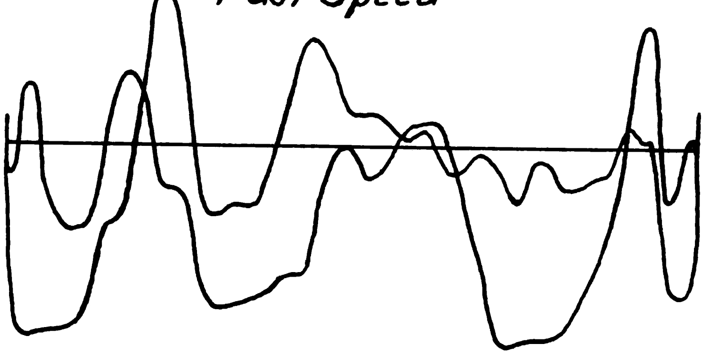




SUCTION PIPE
10# SPRING

Slow Speed

Fast Speed



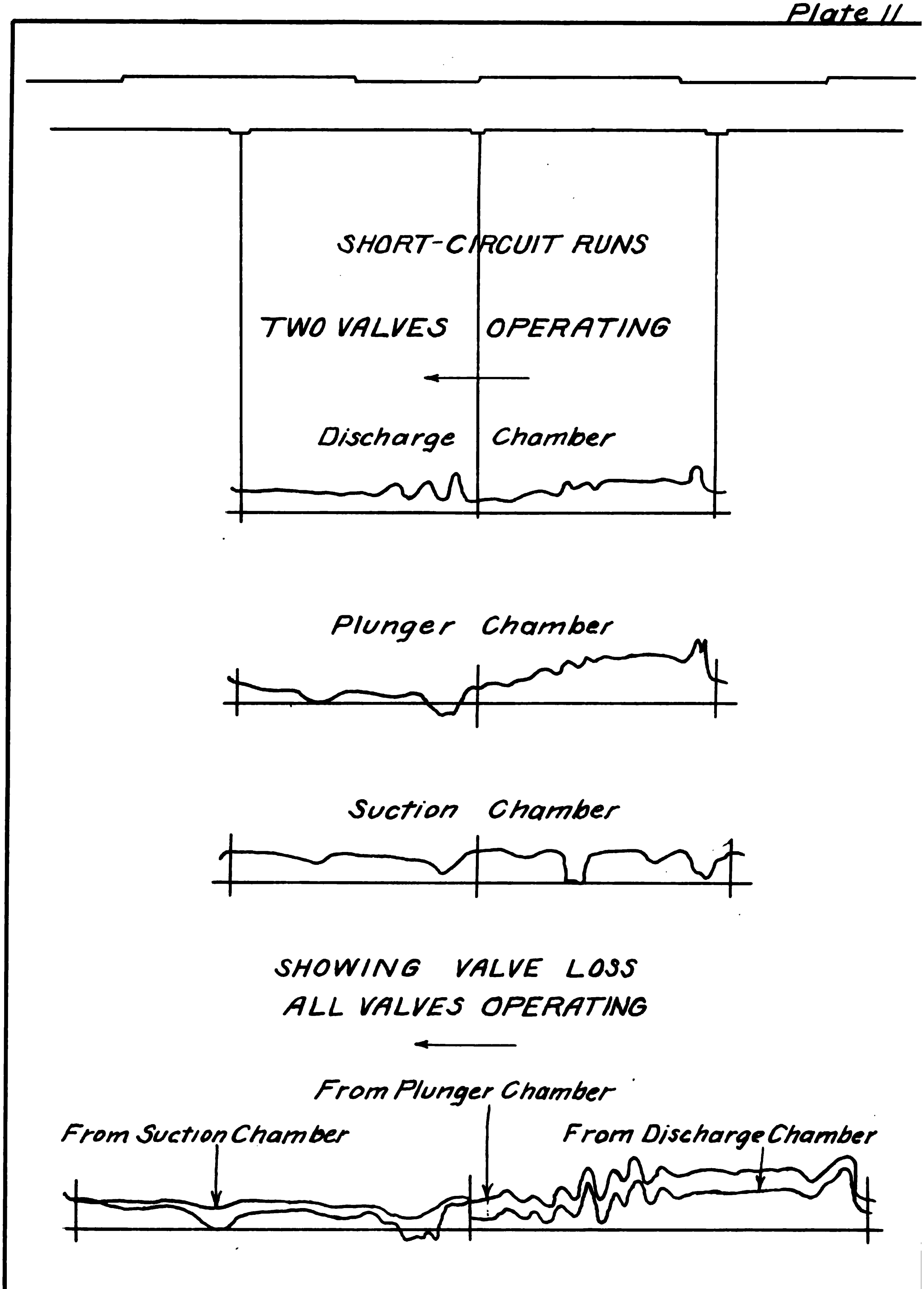

SHORT-CIRCUIT RUNS
TWO VALVES OPERATING
Discharge Chamber
Plunger Chamber
Suction Chamber
SHOWING VALVE LOSS
ALL VALVES OPERATING
From Plunger Chamber
From Suction Chamber
From Discharge Chamber

Plate 12

VALVE LIFT DIAGRAMS — REGULAR RUNS

Three Valves in Each Deck Working

Run A30 Speed = 39 R.P.M.

Run E15 Speed = 67.5 R.P.M.

Valve A1 Valve A2 Valve A3

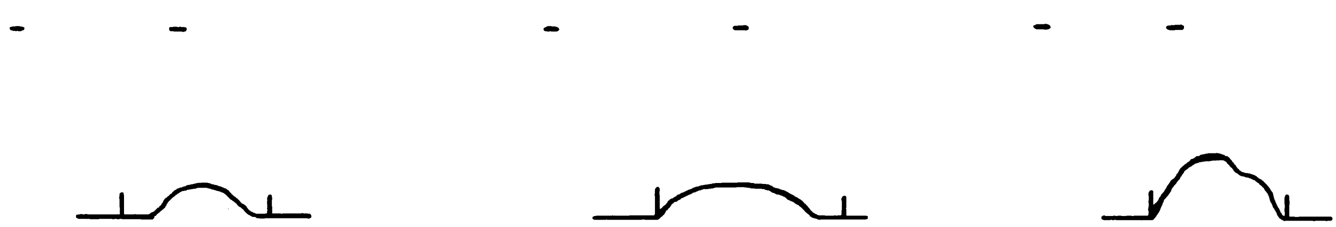
Direction of Propagation ←

Plate 13

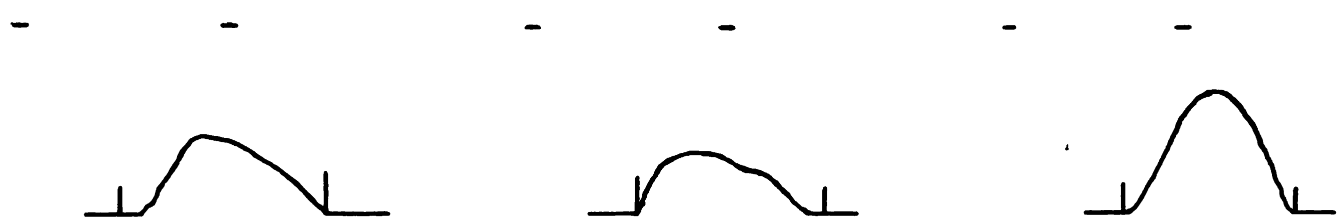
VALVE LIFT DIAGRAMS – REGULAR RUNS

Three Valves in Each Deck Working

Run A15 Speed = 40. R.P.M.



Run E15 Speed = 67.5 R.P.M.



Valve B1 Valve B2 Valve B3

Direction of Propagation ←

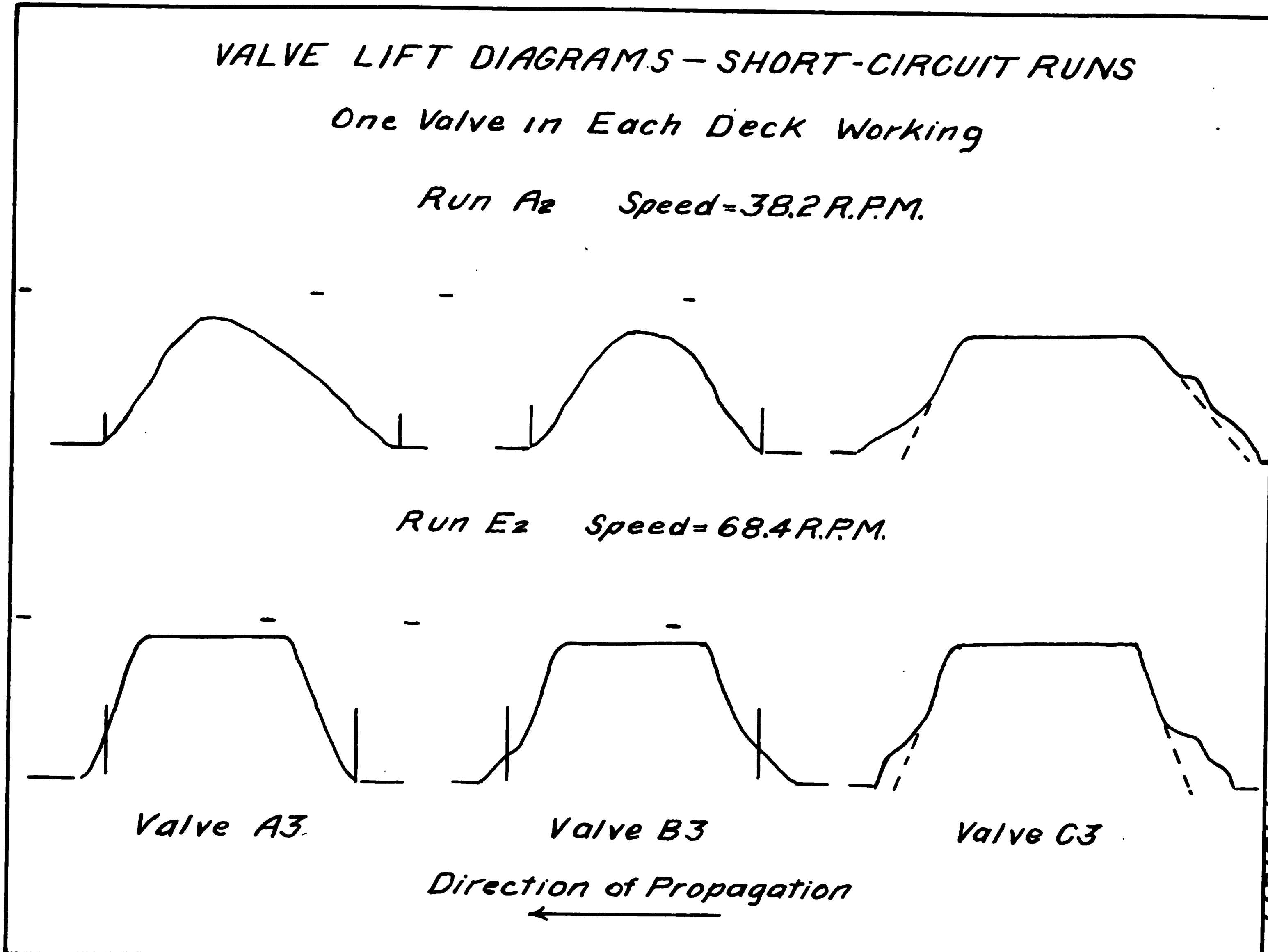
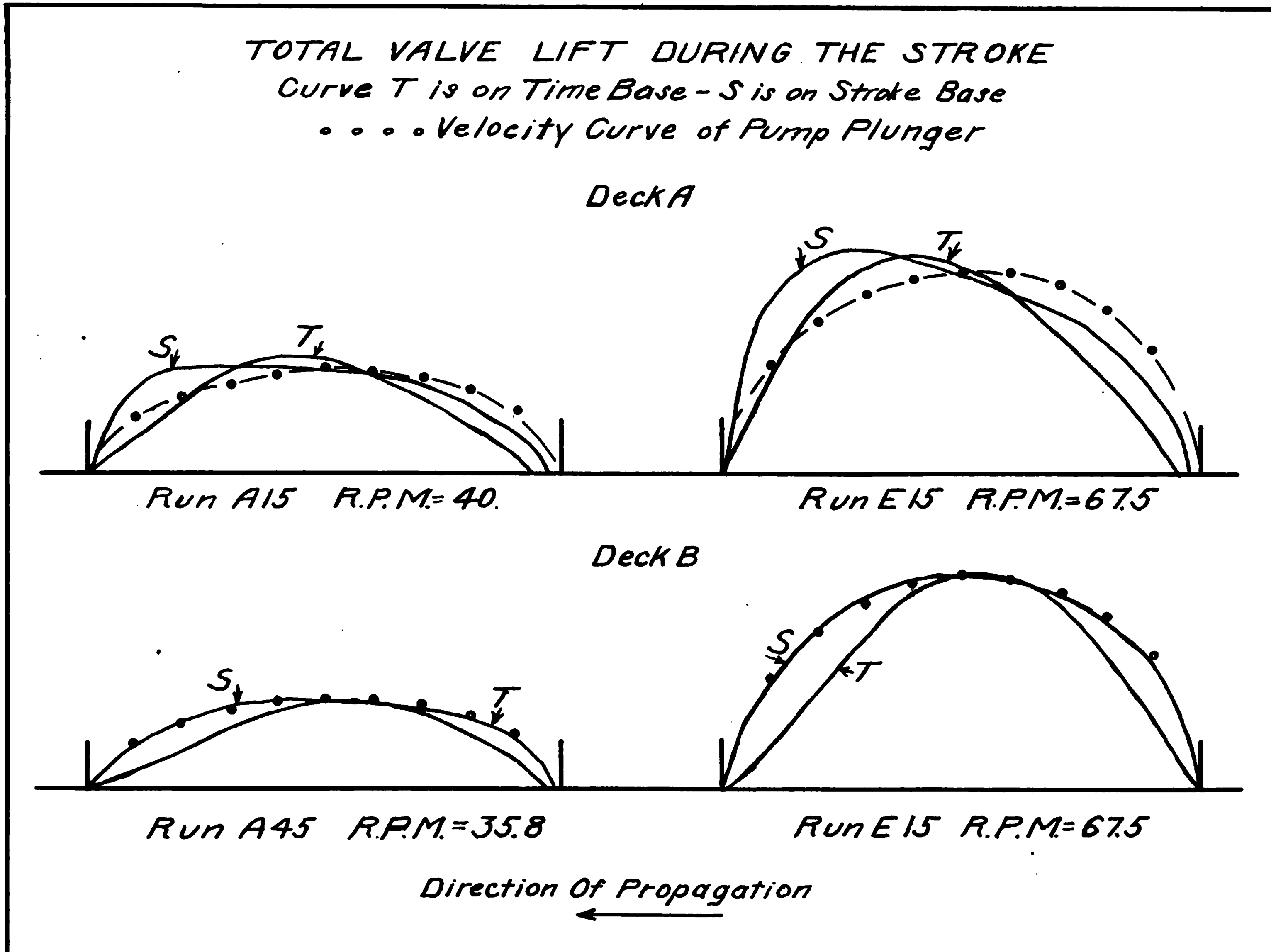
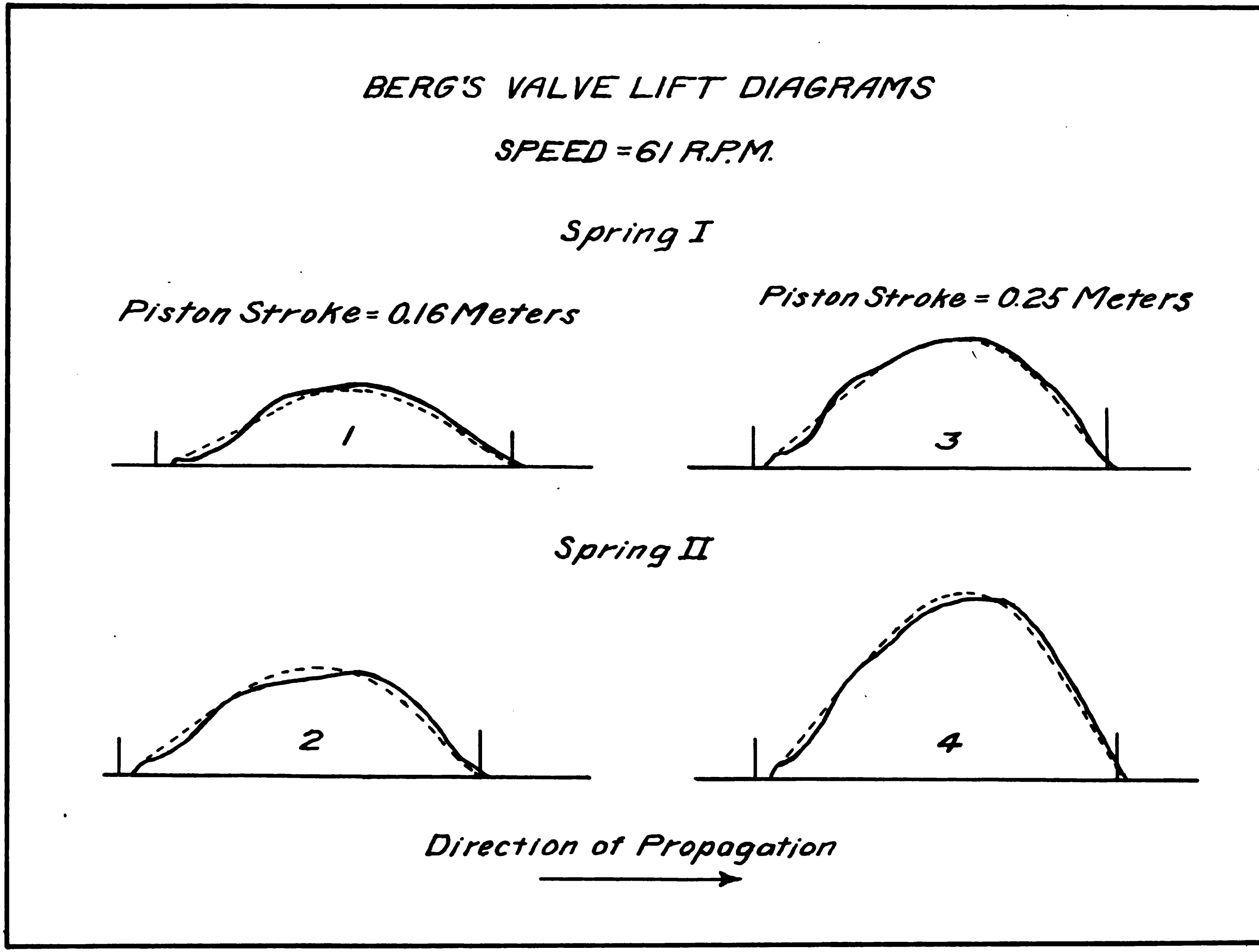

VALVE LIFT DIAGRAMS — SHORT-CIRCUIT RUNS
One Valve in Each Deck Working
Run A2 Speed = 38.2 R.P.M.
Run E2 Speed = 68.4 R.P.M.
Valve A3
Valve B3
Valve C3
Direction of Propagation

Plate 15


TOTAL VALVE LIFT DURING THE STROKE
Curve T is on Time Base - S is on Stroke Base
. . . . Velocity Curve of Pump Plunger
Deck A
S
T
Run A15 R.P.M.= 40.
S
T
Run E15 R.P.M.=67.5
Deck B
S
T
Run A45 R.P.M.=35.8
S
T
Run E15 R.P.M.=67.5
Direction Of Propagation


BERG'S VALVE LIFT DIAGRAMS
SPEED = 61 R.P.M.
Spring I
Piston Stroke = 0.16 Meters
Piston Stroke = 0.25 Meters
1
3
Spring II
2
4
Direction of Propagation

(Sample Log Sheet – Regular Run)

UNIVERSITY OF WISCONSIN – HYDRAULIC LABORATORY

Experiment on Fairbanks-Morse Pump

Data by Corp, North, McCollister – Computed by C. N. McC.

Date of Experiment Apr. 6, 1911.

General Data. Run A 60 (Slow Speed)

No.	Time	Amp.	Volts	Elec. H.P.	Pump H.P.	Rev.	R.P.M.	Dis. P. Lbs. per Sq. In.	Suc. P. Ft. Merc.	Float Gage Ft.	Valve Lift Ins.
1	9-30	21.0	470			2331					
	9-35	20.5	475			507	35.2	60	-.45	4.49	
2	9-40	20.5	475			2685					
	9-45	20.5	480			862	35.4	60	-.45	4.49	
3	9-50	21.0	485			3043					
	9-55	21.0	485			3226	36.6	60	-.45	4.49	D_1 .07
4	10-00	21.0	490			3410					D_2 .06
	10-05	20.5	485			3594	36.8	60	-.45	4.49	D_3 .46
Ave.		20.75	480.6				36.1	60			
Cor.		21.20	479.0	13.6	9.8			59			

Amp.	Volts	R.P.M.	Brake Load	Elec. H.P.	Brake H.P.	Amp.	Volts	R.P.M.	Brake Load	Elec. H.P.	Brake H.P.
23.0	477		95			17.5	490		65		
23.5	480					17.5	495				
23.0	485					18.0	485				
23.5	485					18.0	485				
23.5	485	181				17.5	485	186			
23.5	485					17.5	480				
23.5	485					18.0	485				
23.5	485					18.0	485				
23.5	482					17.5	483				
23.0	480	182				18.0	485	185			
23.35	482.9	181.5	Ave.			17.75	485.8	185.5	Ave.		
23.90	481.0		Cor.	15.4	11.5	18.10	484.0		Cor.	11.8	8.03

20.5	485		80		
20.5	485				
20.5	483				
20.5	485				
20.5	485	184			
20.5	483				
20.5	485				
20.5	485				
20.5	489				
20.5	489	185			
20.5	484.5	184.5	Ave.		
20.9	483.0		Cor.	13.5	9.81

Time	Weir Hook Gage Ft.	Time	Weir Hook Gage Ft.	Dis-charge C.F.S.
9-30	.3311	9-52½	.3410	
9-32½	.3322	9-55	.3399	
9-35	.3332	9-57½	.3404	
9-37½	.3337	10-00	.3405	
9-40	.3337	10-02½	.3405	
9-42½	.3337	10-05	.3425	
9-45	.3358	Ave.	.3371	
9-47½	.3373	Cor.	.3257	.364
9-50	.3402			

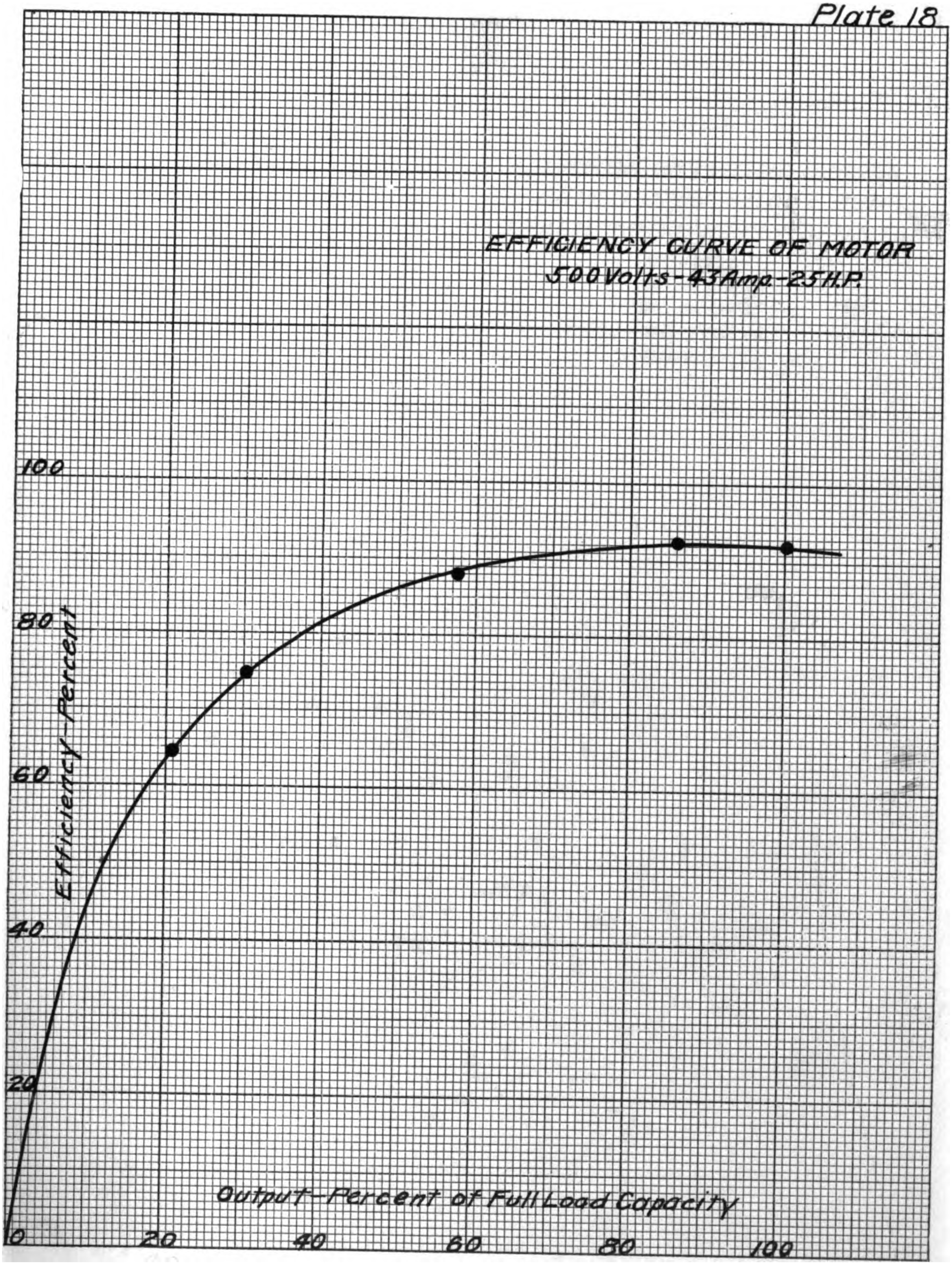
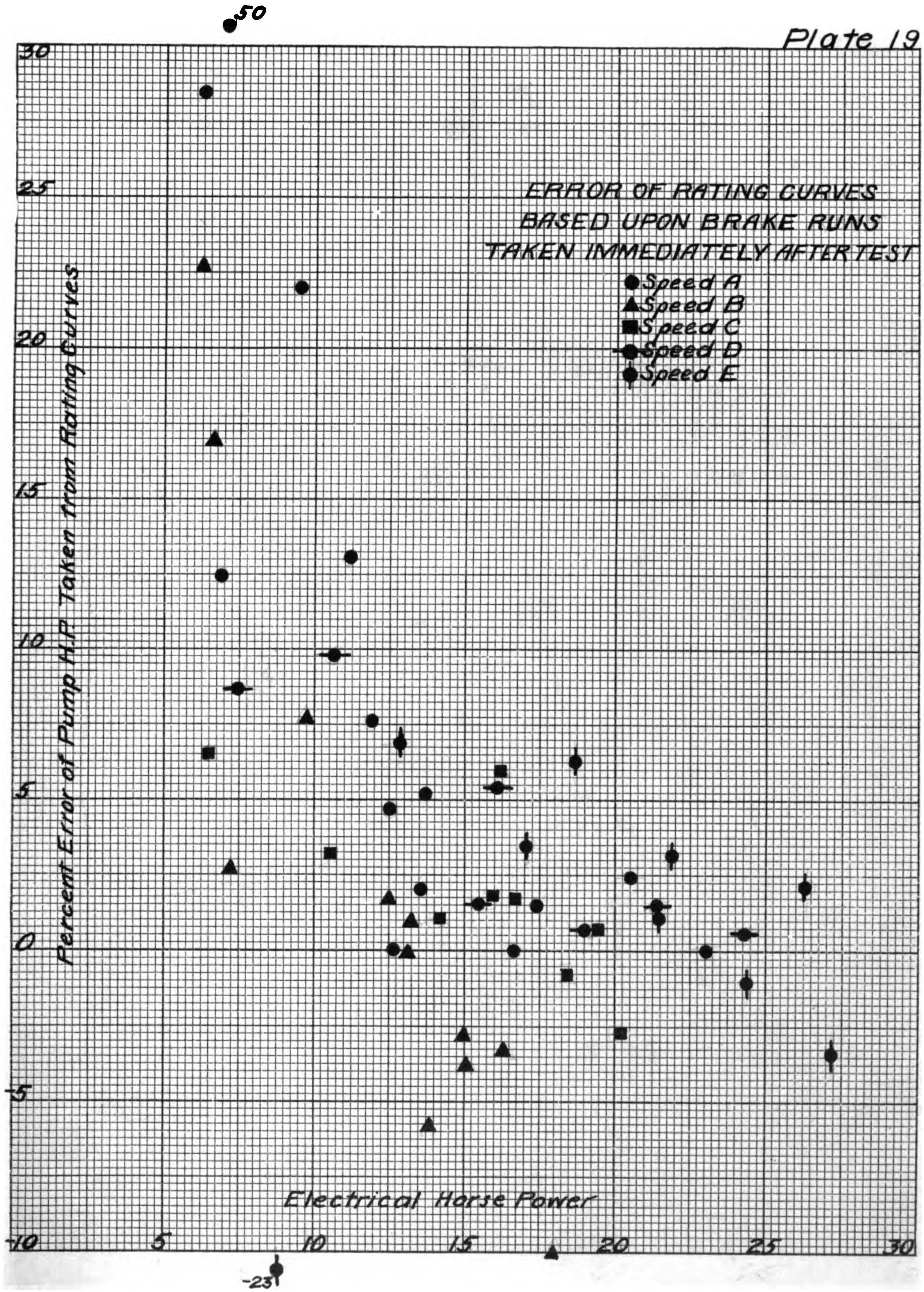
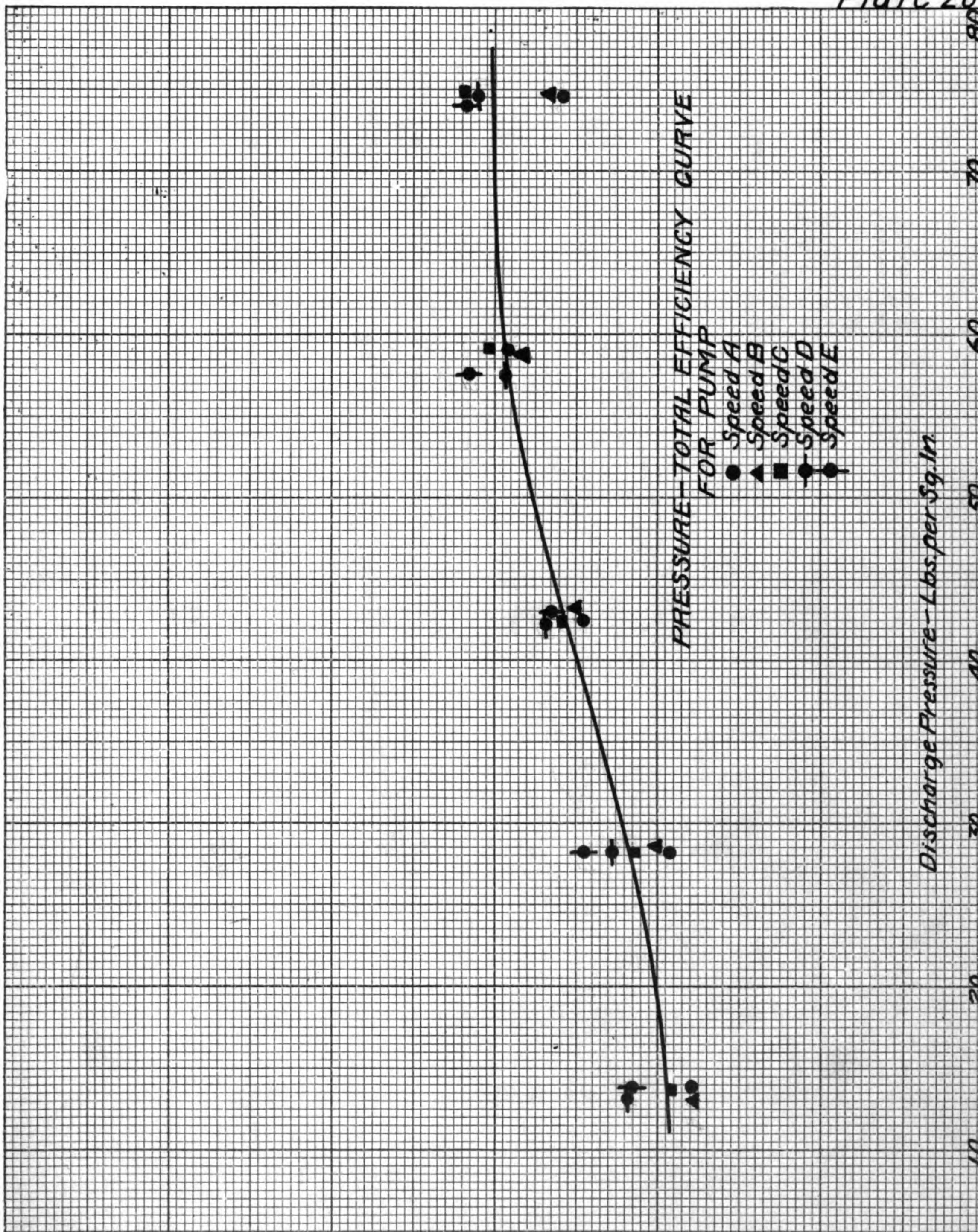

EFFICIENCY CURVE OF MOTOR
500 Volts - 43 Amp - 25 H.P.
100
80
60
40
20
0
Efficiency - Percent
Output - Percent of Full Load Capacity
20
40
60
80
100

Plate 19



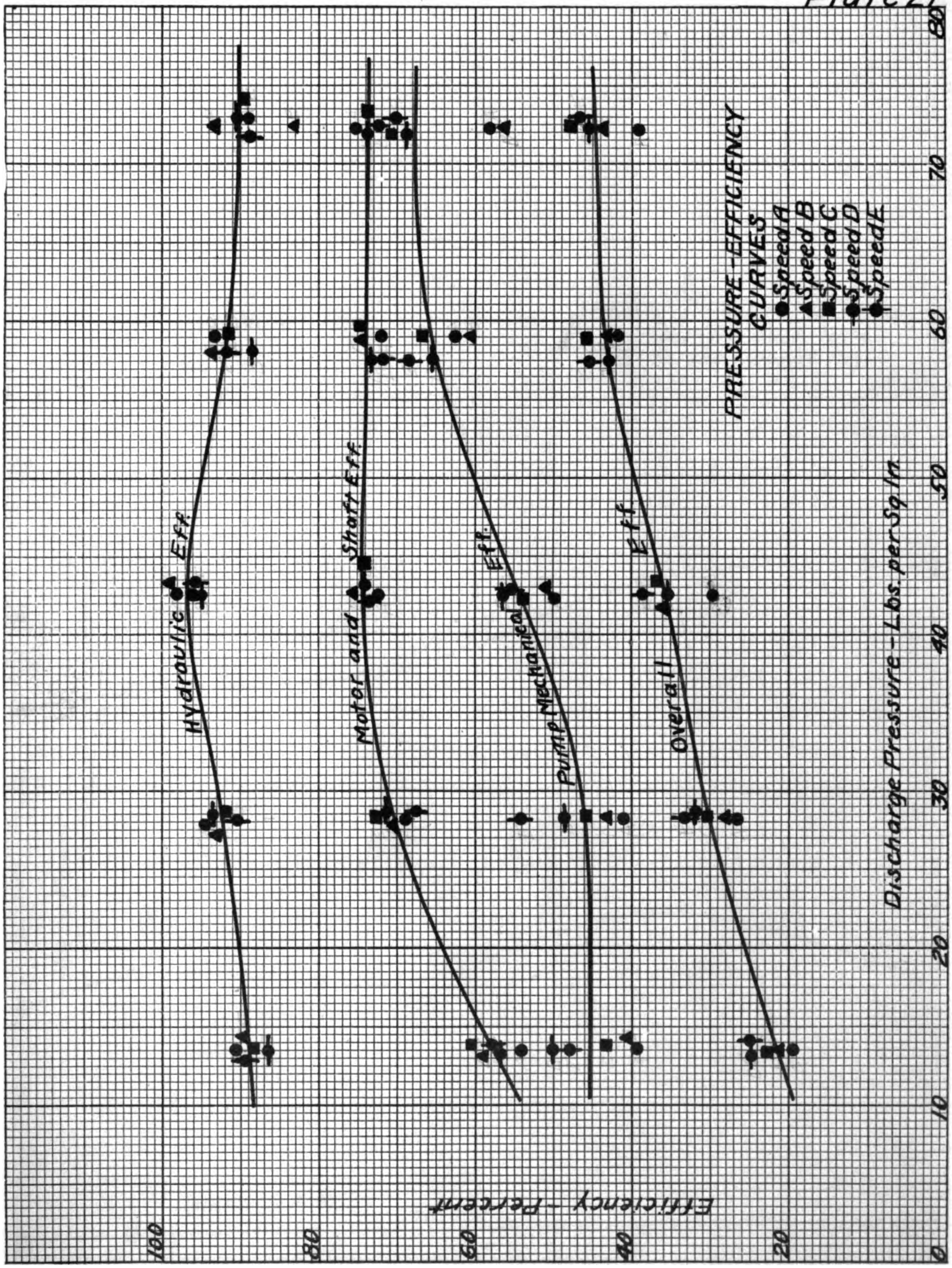
ERROR OF RATING CURVES
BASED UPON BRAKE RUNS
TAKEN IMMEDIATELY AFTER TEST
Speed A
Speed B
Speed C
Speed D
Speed E
Percent Error of Pump H.P. Taken from Rating Curves
Electrical Horse Power
30
25
20
15
10
5
0
-5
-10
5
10
15
20
25
30
50
-23

Plate 20



PRESSURE-TOTAL EFFICIENCY CURVE
FOR PUMP
Speed A
Speed B
Speed C
Speed D
Speed E
Discharge Pressure-Lbs. per Sq. In.
Efficiency-Percent
0
10
20
30
40
50
60
70
80
20
40
60
80
100

Plate 21


PRESSURE-EFFICIENCY CURVES
Speed A
Speed B
Speed C
Speed D
Speed E
Hydraulic Eff.
Motor and Shaft Eff.
Pump Mechanical Eff.
Over all Eff.
Discharge Pressure - Lbs. per Sq. In.
Efficiency - Percent
0
10
20
30
40
50
60
70
80
100

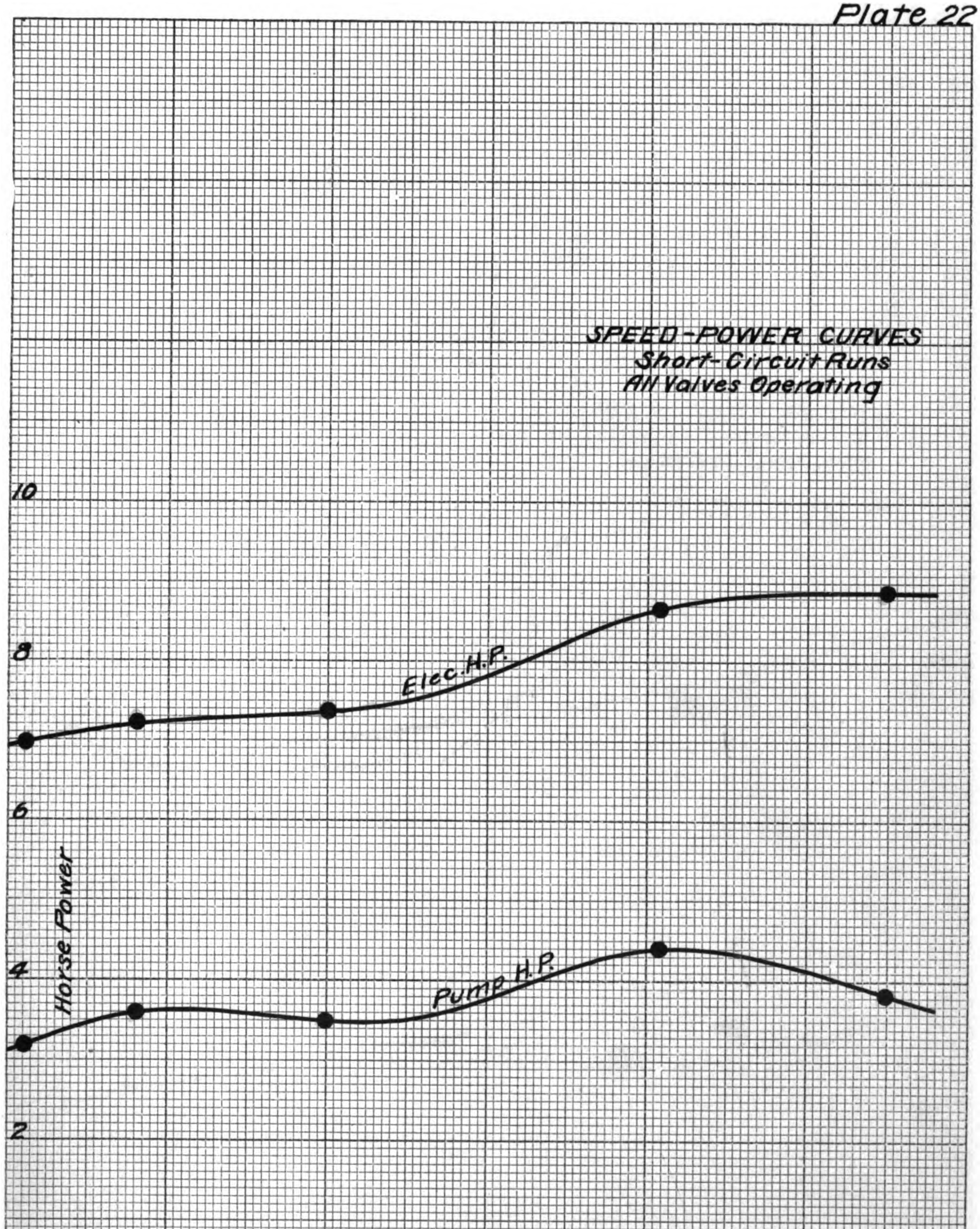
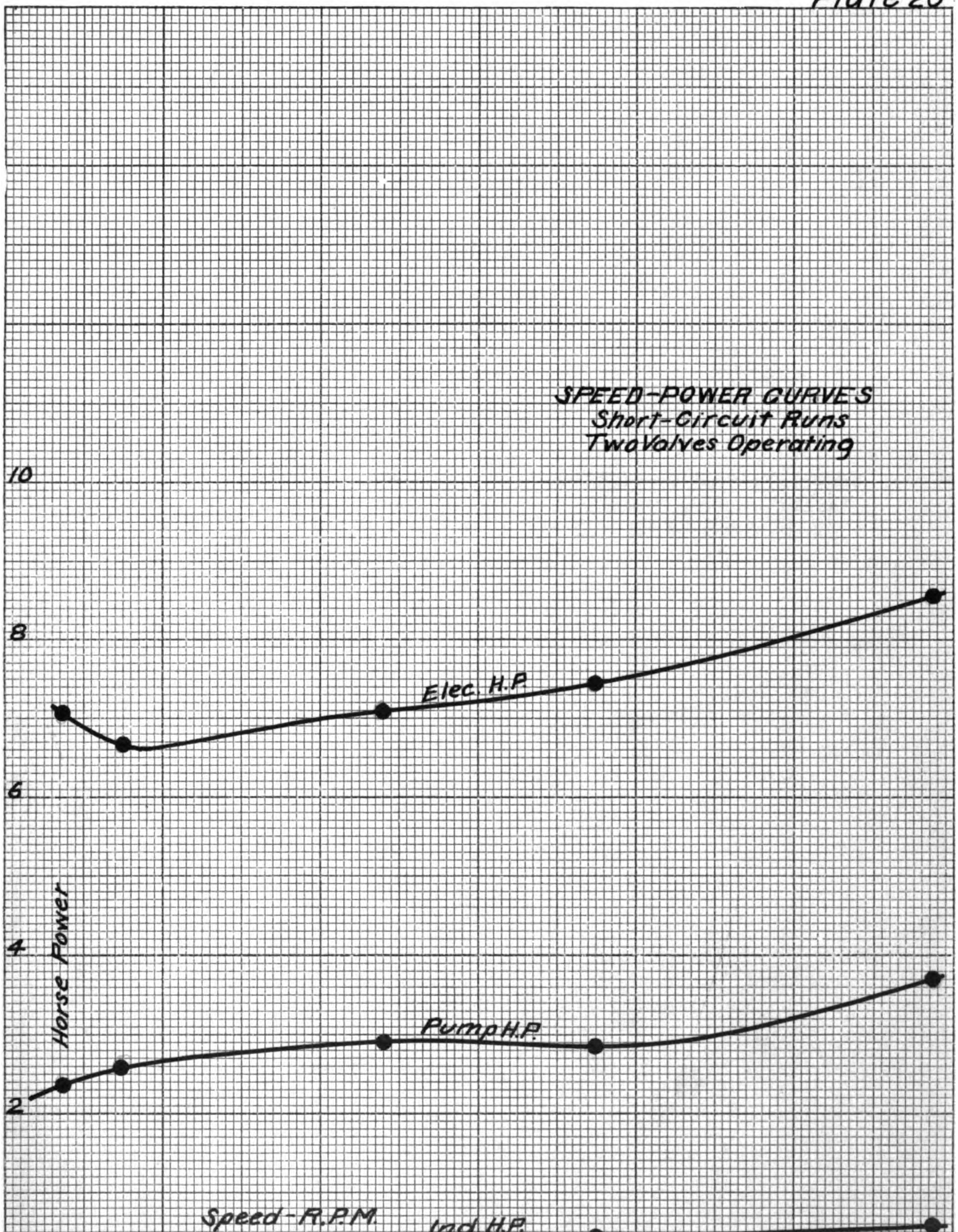

SPEED-POWER CURVES
Short-Circuit Runs
All Valves Operating
Elec. H.P.
Pump H.P.
Ind. H.P.
Speed-R.P.M.
Horse Power
10
8
6
4
2
40
45
50
55
60
65
70

Plate 23



SPEED-POWER CURVES
Short-Circuit Runs
Two Valves Operating
Elec. H.P.
Pump H.P.
Speed-R.P.M.
Ind. H.P.
Horse Power
10
8
6
4
2
40
45
50
55
60
65
70

Plate 24

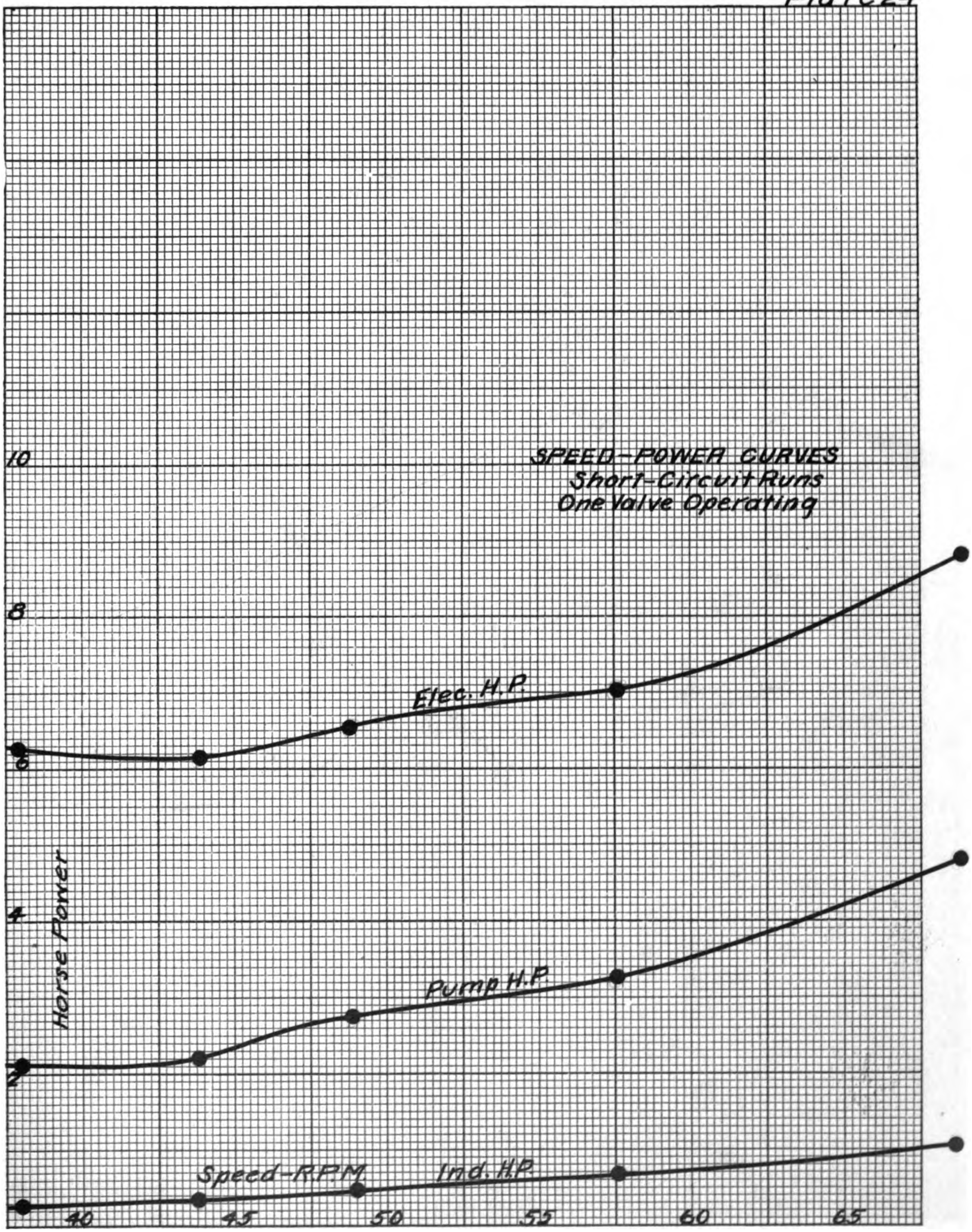

SPEED-POWER CURVES
Short-Circuit Runs
One Valve Operating
Elec. H.P.
Pump H.P.
Ind. H.P.
Speed-R.P.M.
Horse Power
10
8
6
4
2
40
45
50
55
60
65

Plate 25

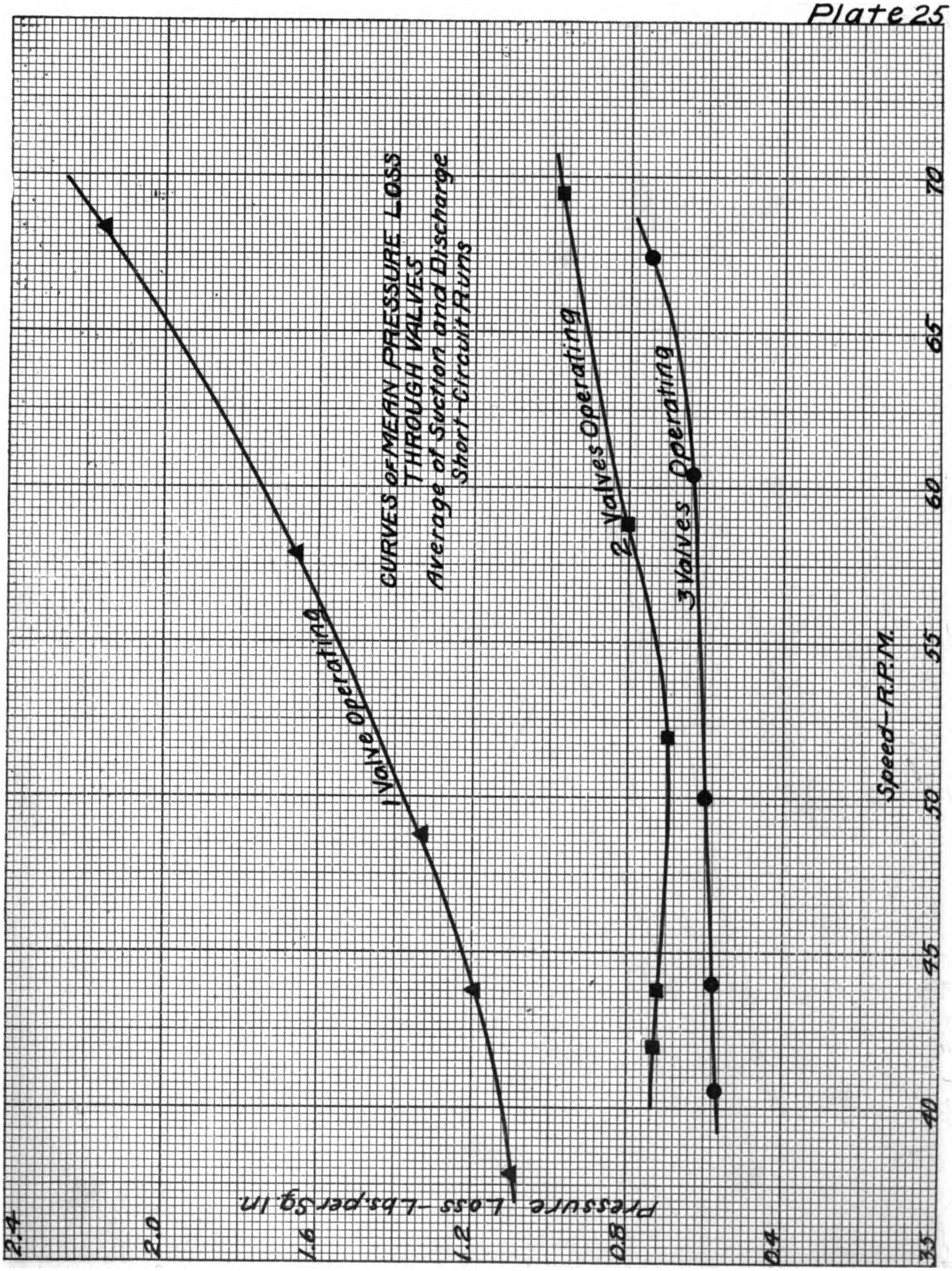
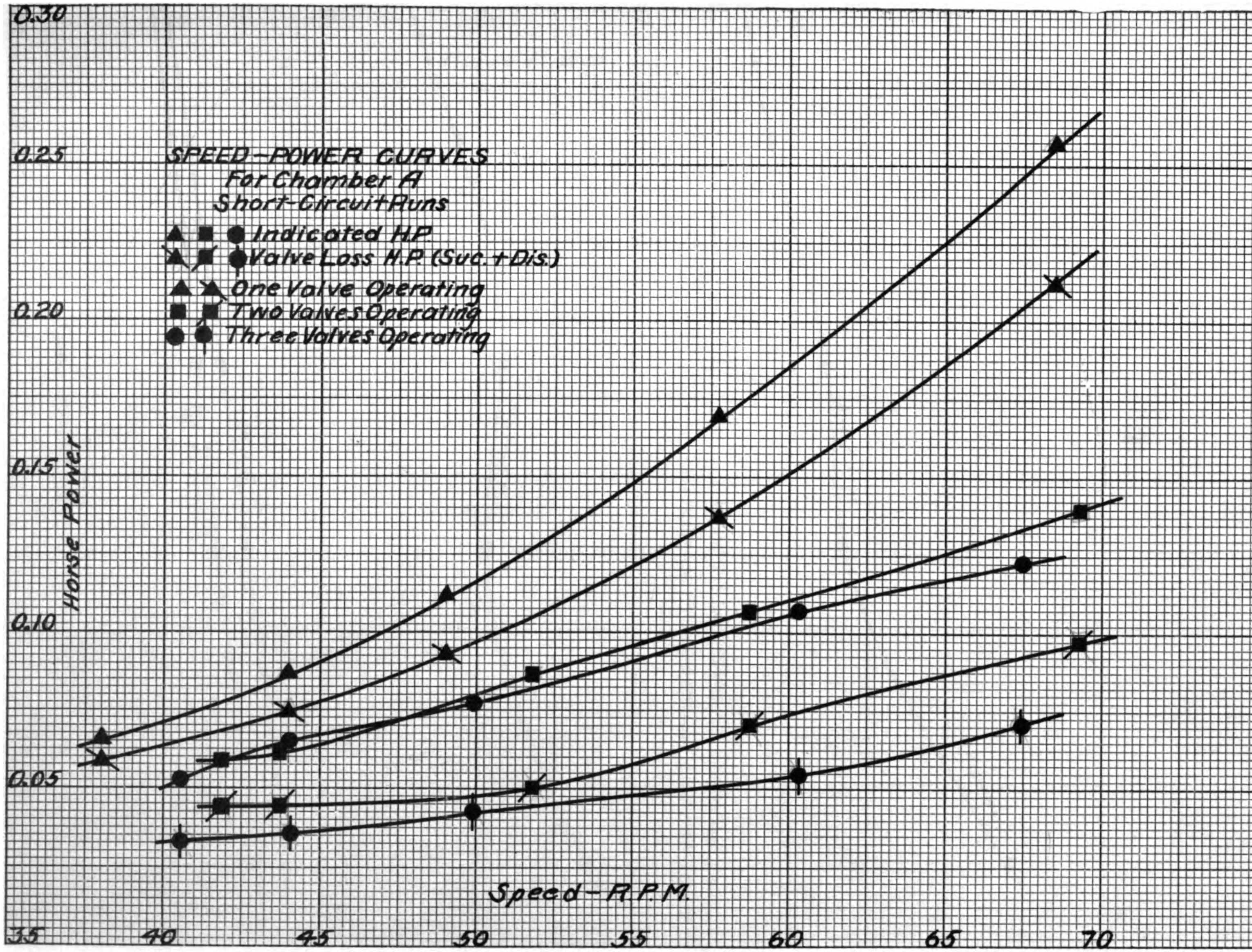
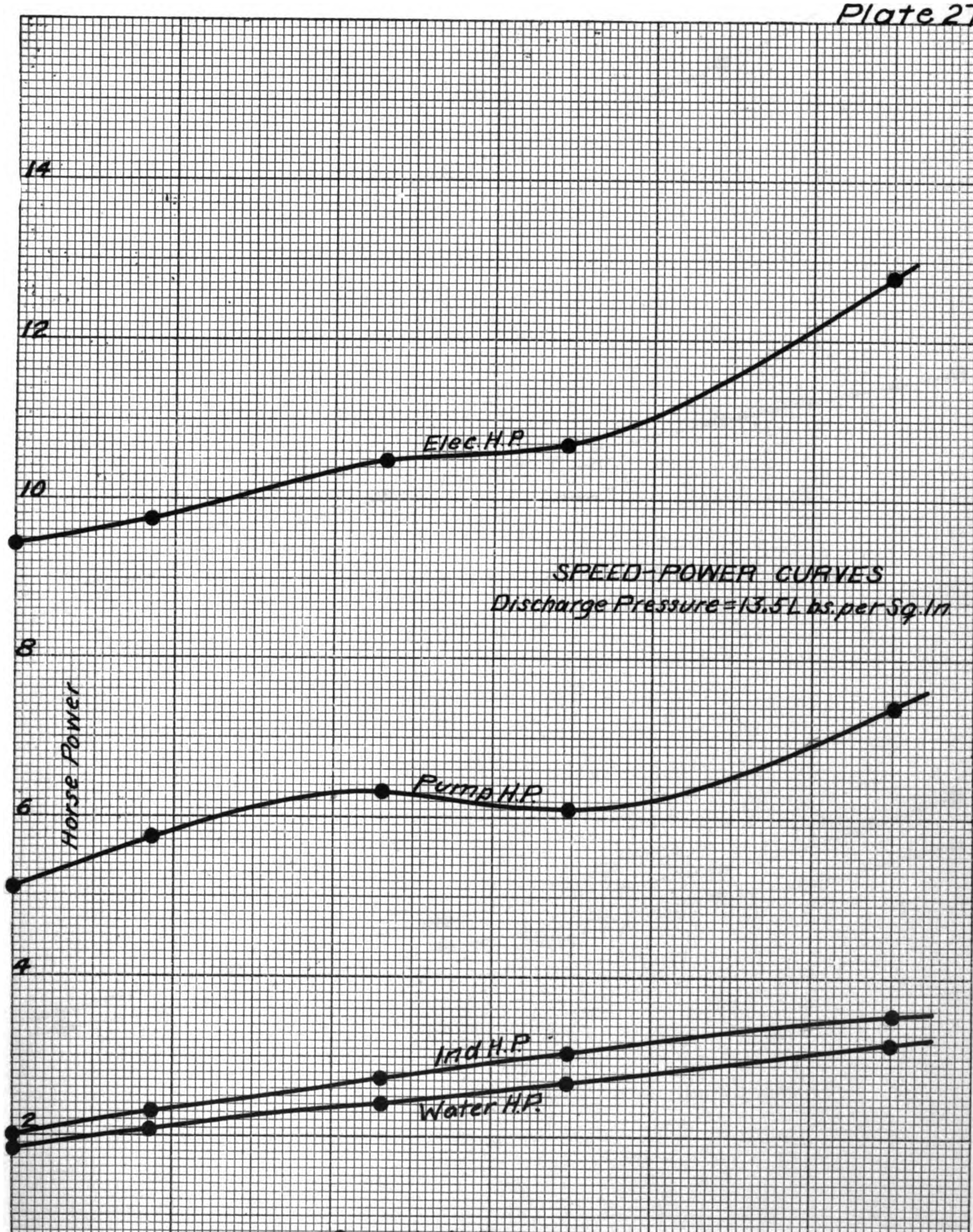

CURVES OF MEAN PRESSURE LOSS THROUGH VALVES
Average of Suction and Discharge
Short-Circuit Runs
1 Valve Operating
2 Valves Operating
3 Valves Operating
Speed—R.P.M.
Pressure Loss—Lbs. per Sq. In.
35
40
45
50
55
60
65
70
0.4
0.8
1.2
1.6
2.0
2.4

Plate 26


SPEED-POWER CURVES
For Chamber A
Short-Circuit Runs
Indicated H.P.
Valve Loss H.P. (Suc. + Dis.)
One Valve Operating
Two Valves Operating
Three Valves Operating
Horse Power
0.30
0.25
0.20
0.15
0.10
0.05
Speed-R.P.M.
35
40
45
50
55
60
65
70


SPEED-POWER CURVES
Discharge Pressure = 13.5 Lbs. per Sq. In.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Speed - R.P.M.
14
12
10
8
6
4
2
40
45
50
55
60
65
70

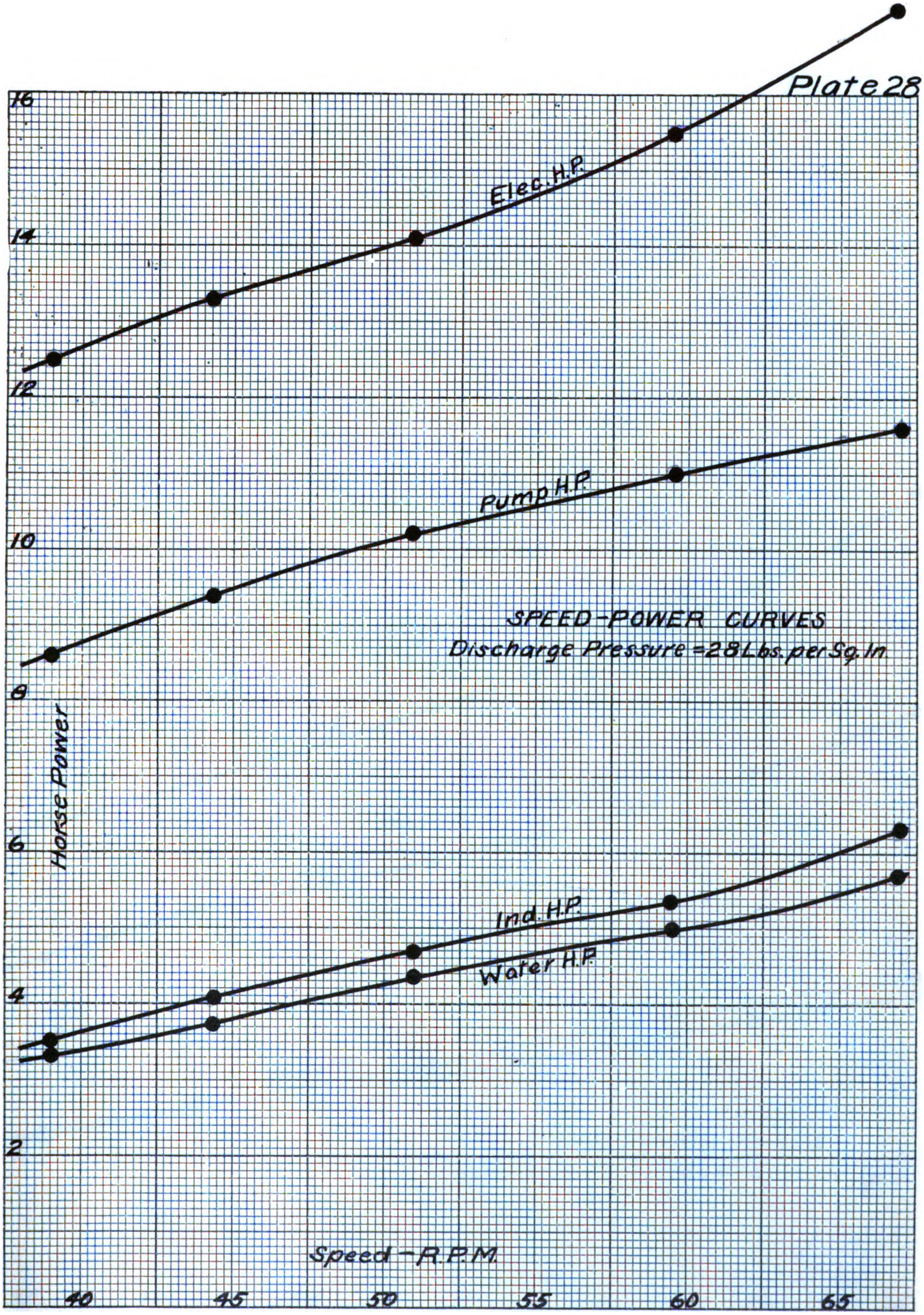
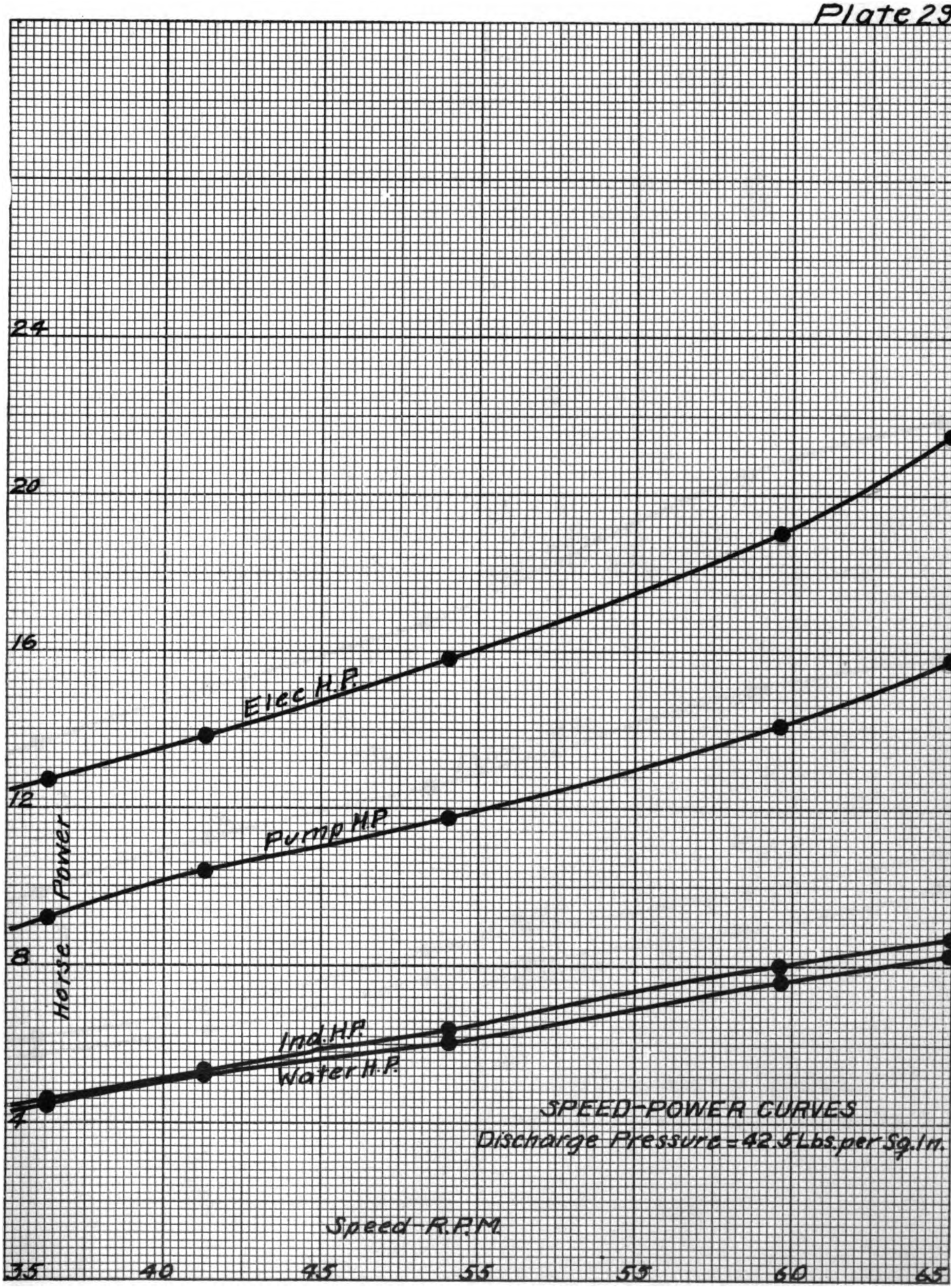


Plate 28
SPEED-POWER CURVES
Discharge Pressure =28 Lbs. per Sq. In.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Speed - R.P.M.
16
14
12
10
8
6
4
2
40
45
50
55
60
65

Plate 29



24
20
16
12
8
4
Horse Power
Elec H.P.
Pump H.P.
Ind. H.P.
Water H.P.
SPEED-POWER CURVES
Discharge Pressure = 42.5 Lbs. per Sq. In.
Speed-R.P.M.
35
40
45
55
55
60
65

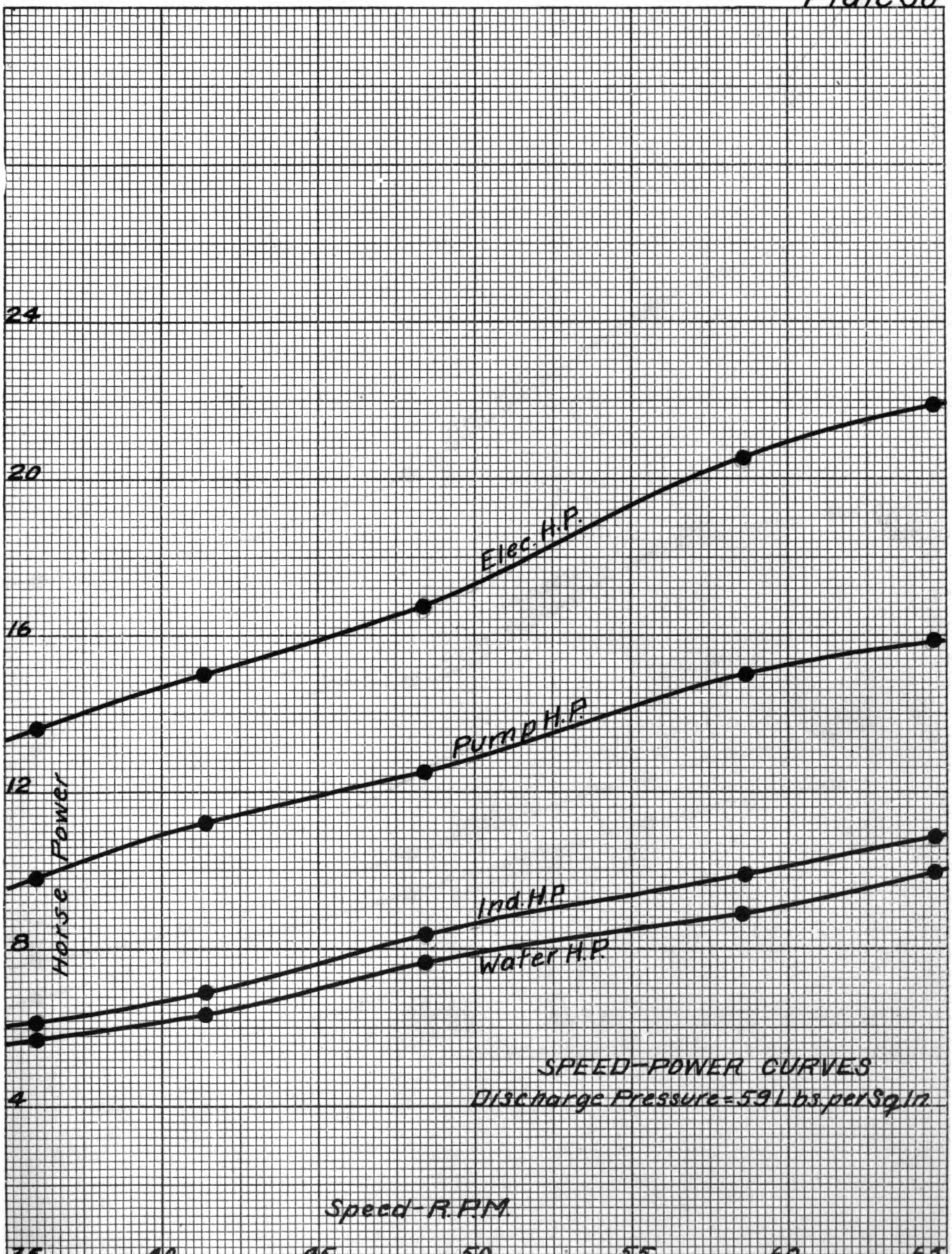
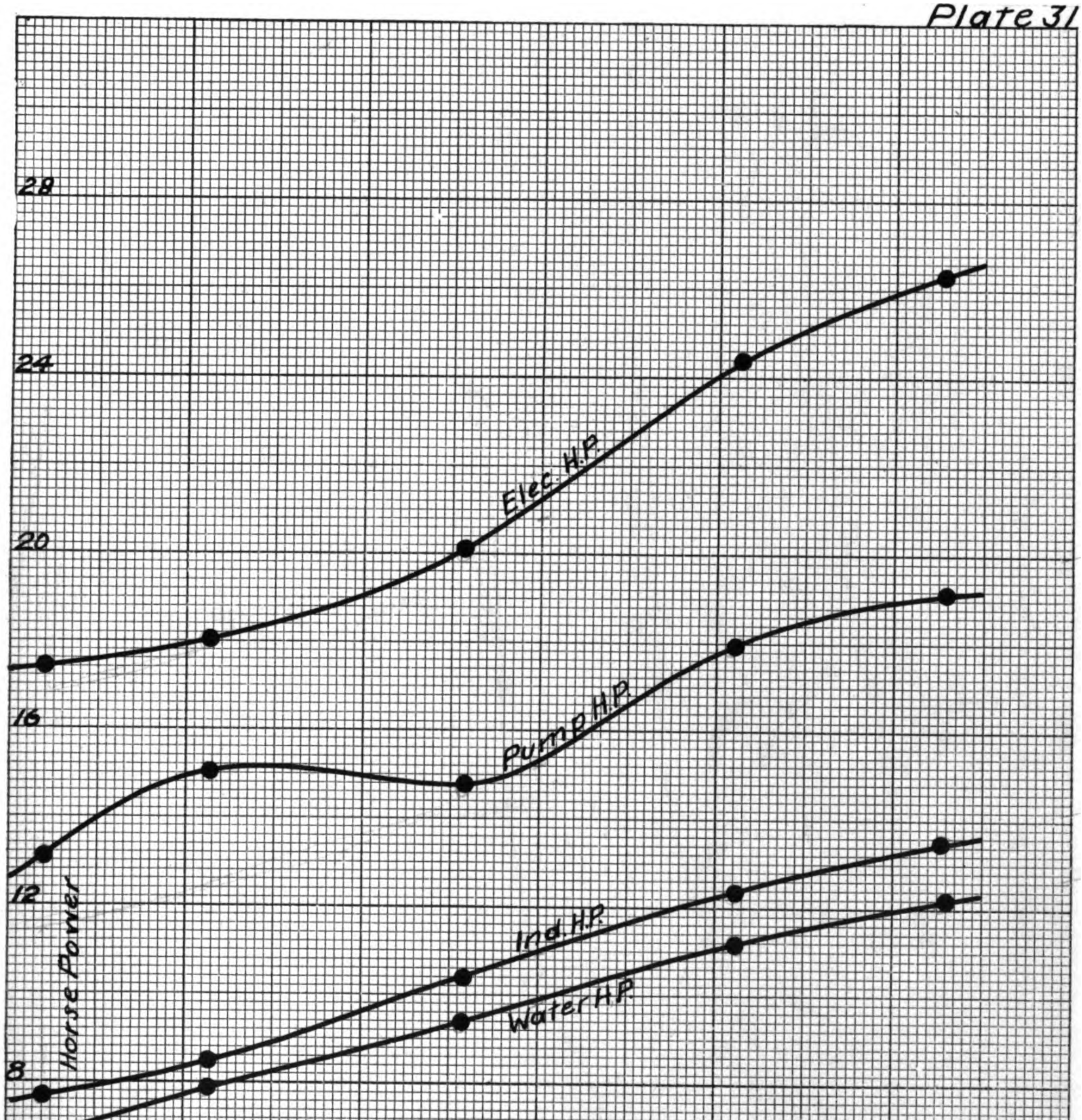
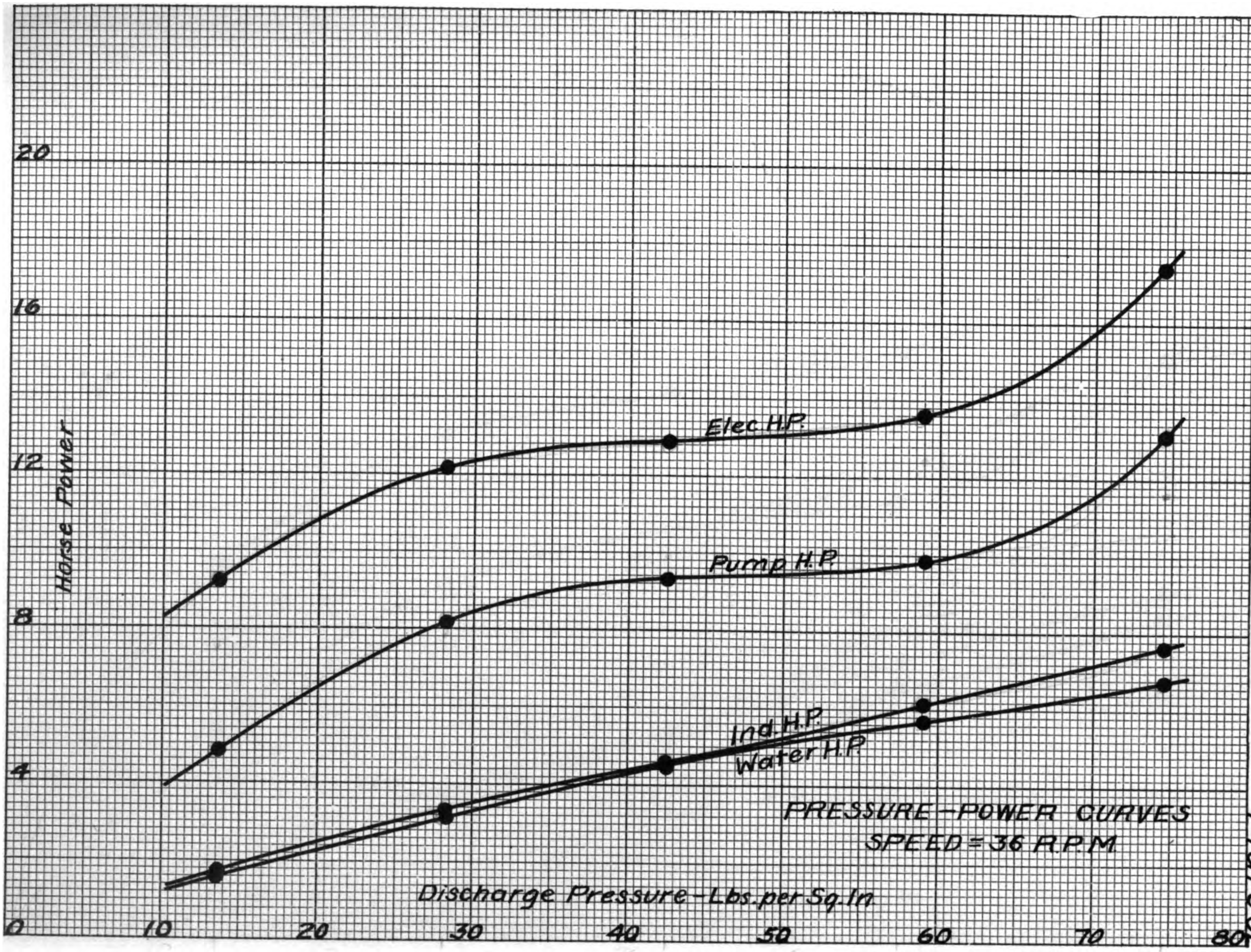

Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
24
20
16
12
8
4
SPEED-POWER CURVES
Discharge Pressure = 59 Lbs. per Sq. In.
Speed-R.P.M.
35
40
45
50
55
60
65

Plate 31



28
24
20
16
12
8
4
Horse Power
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
SPEED-POWER CURVES
Discharge Pressure = 74.5 Lbs. per Sq. In.
Speed - R.P.M.
35
40
45
50
55
60
65

Plate 32



PRESSURE-POWER CURVES
SPEED = 36 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Discharge Pressure - Lbs. per Sq. In.
0
4
8
12
16
20
10
20
30
40
50
60
70
80

Plate 33

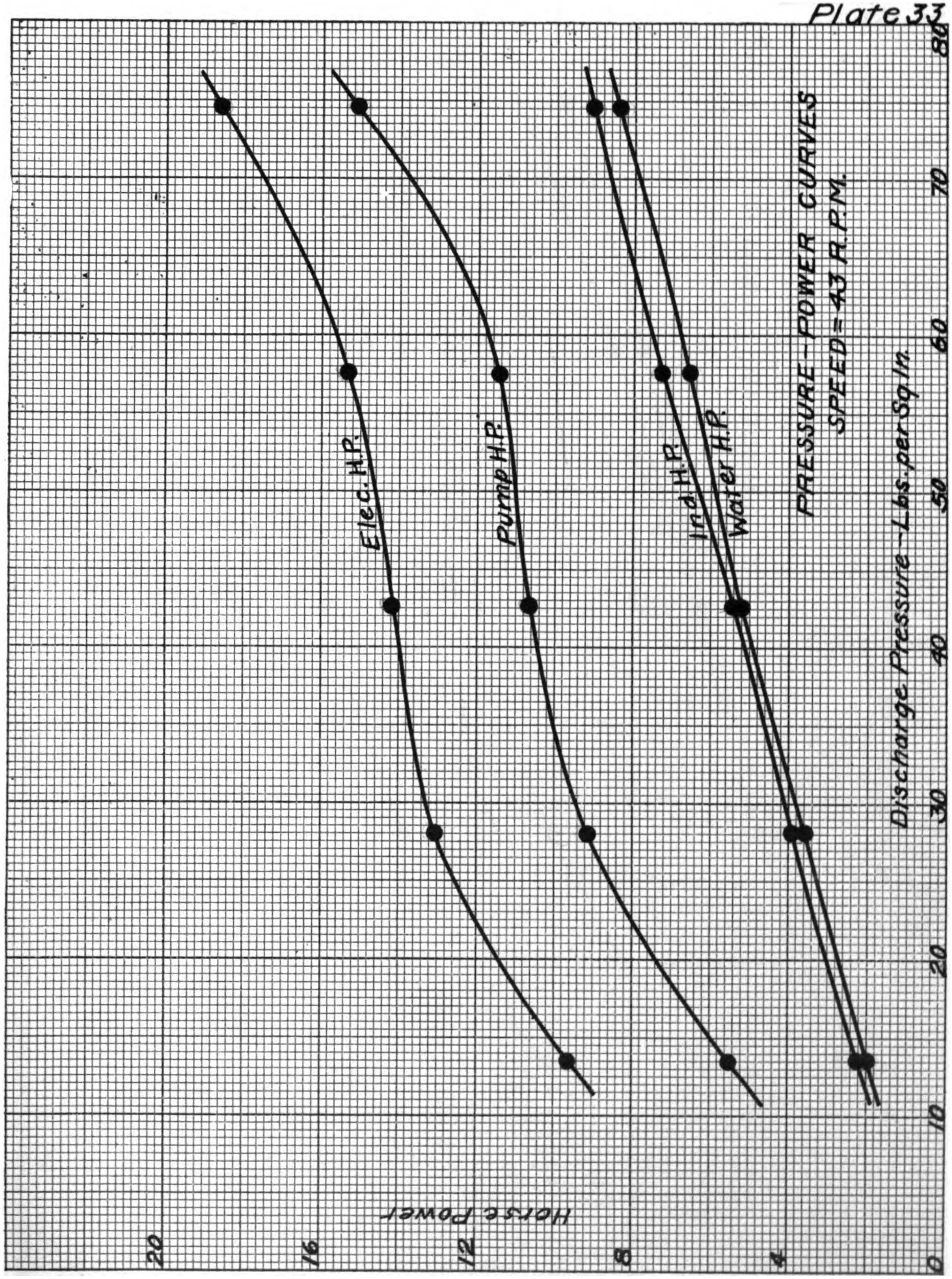

PRESSURE-POWER CURVES
SPEED=43 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Discharge Pressure-Lbs. per Sq. In.
0
10
20
30
40
50
60
70
80
4
8
12
16
20

Plate 34

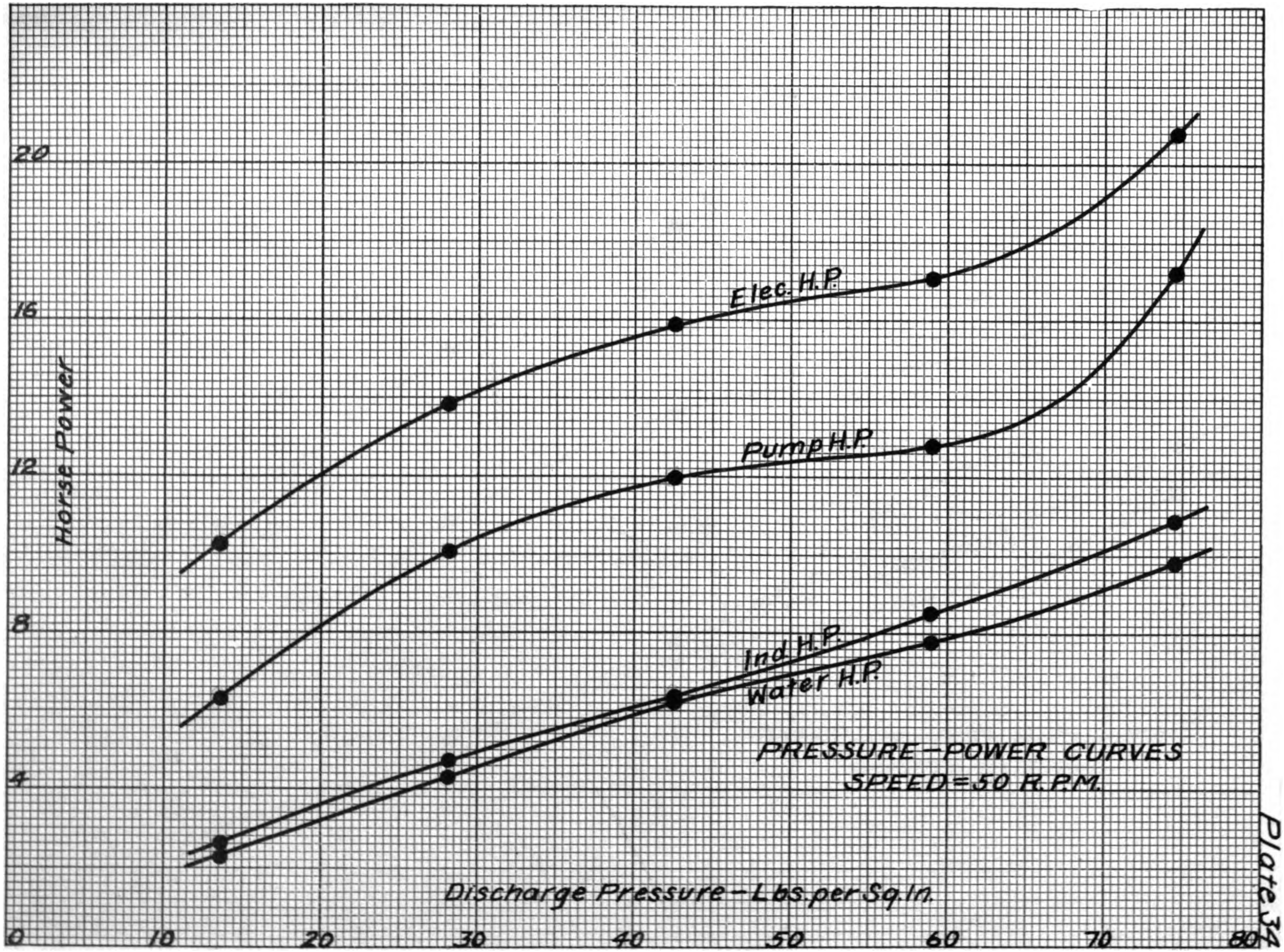
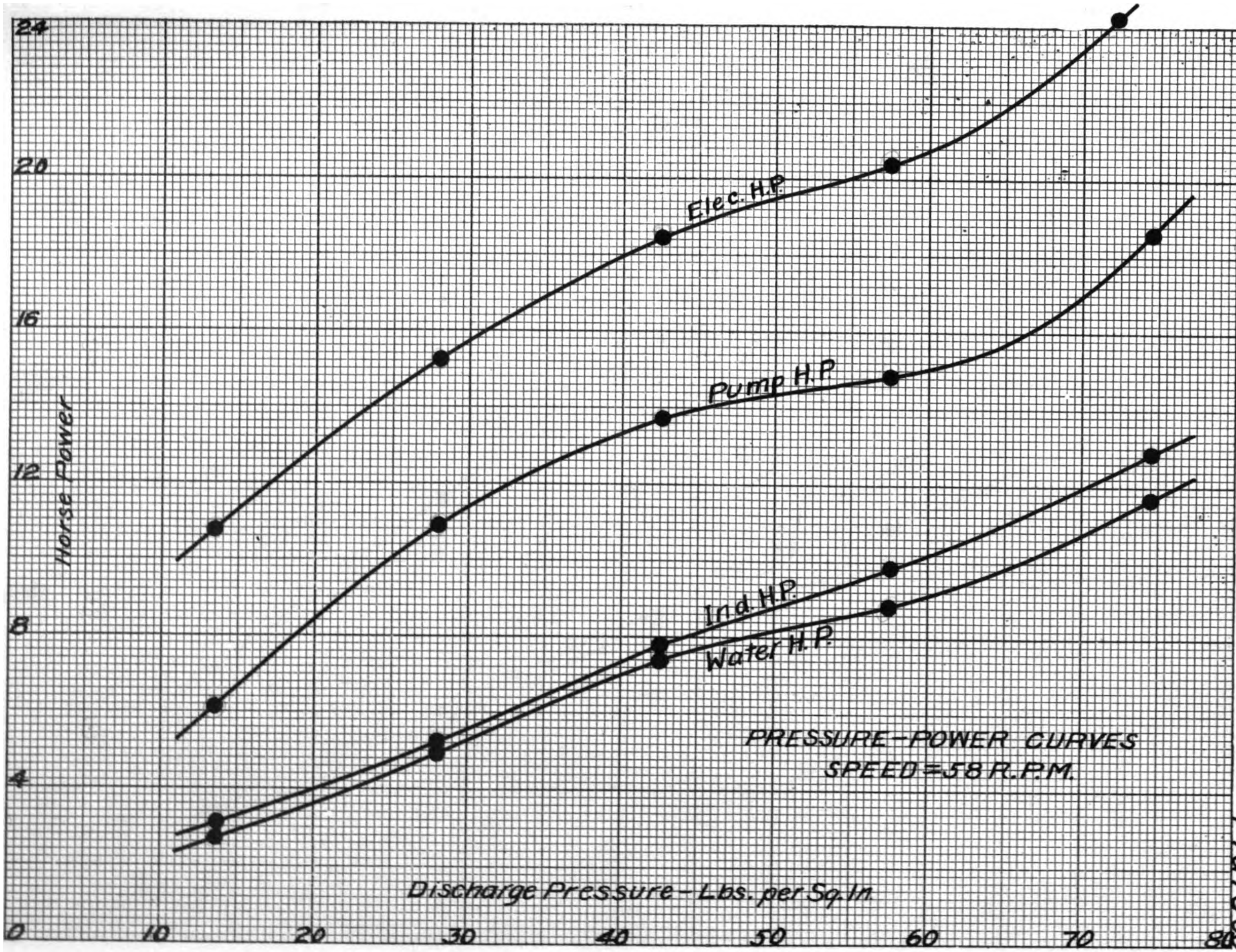
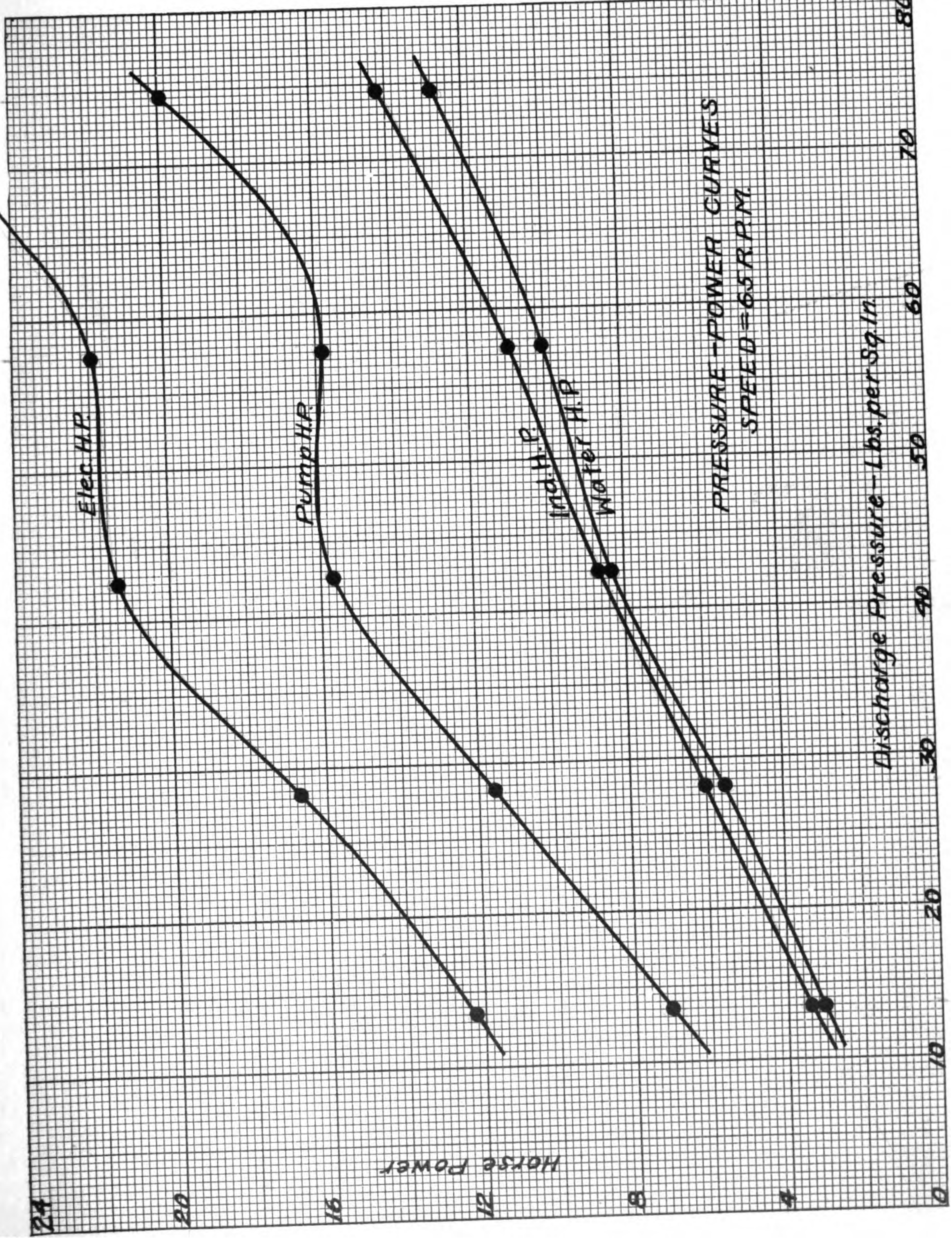

PRESSURE-POWER CURVES
SPEED=50 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Discharge Pressure-Lbs. per Sq. In.
0
10
20
30
40
50
60
70
80
4
8
12
16
20

Plate 35



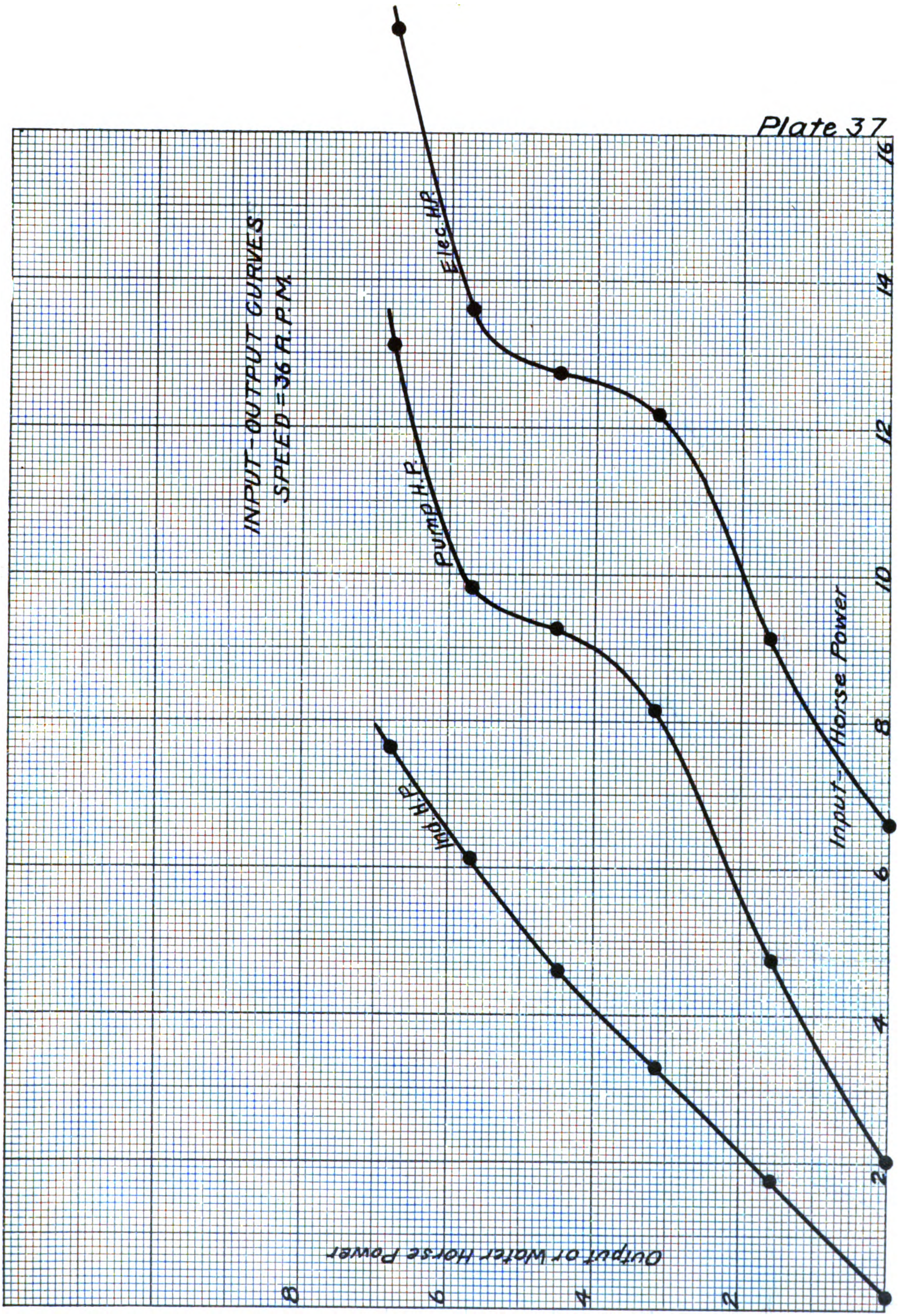
PRESSURE-POWER CURVES
SPEED=58 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Discharge Pressure-Lbs. per Sq. In.
0
10
20
30
40
50
60
70
80
4
8
12
16
20
24

Plate 36



PRESSURE-POWER CURVES
SPEED=65 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Water H.P.
Horse Power
Discharge Pressure-Lbs. per Sq. In.
0
10
20
30
40
50
60
70
80
4
8
12
16
20
24

Plate 37



INPUT-OUTPUT CURVES
SPEED = 36 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Input-Horse Power
Output or Water Horse Power
2
4
6
8
10
12
14
16

Plate 38

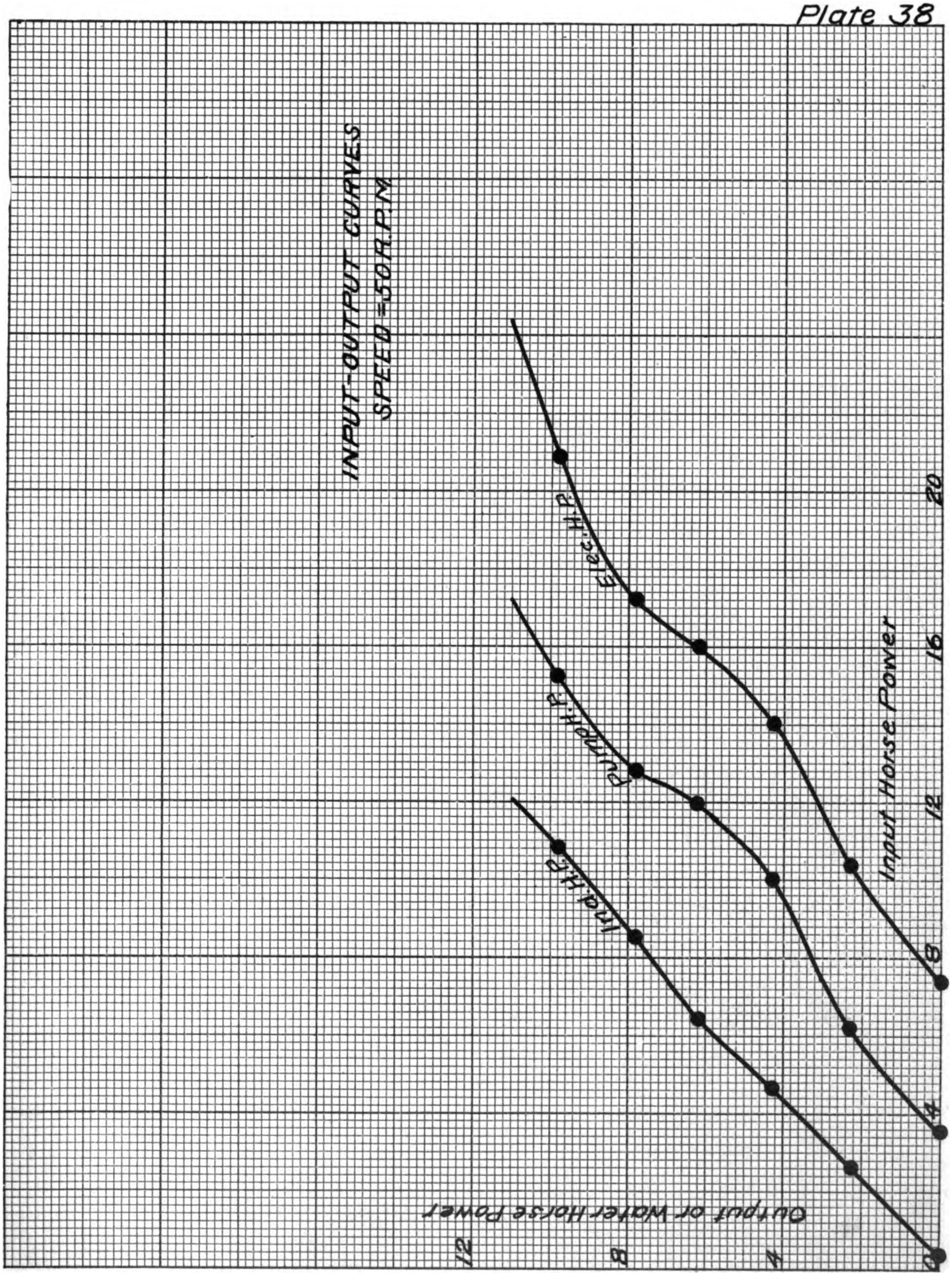
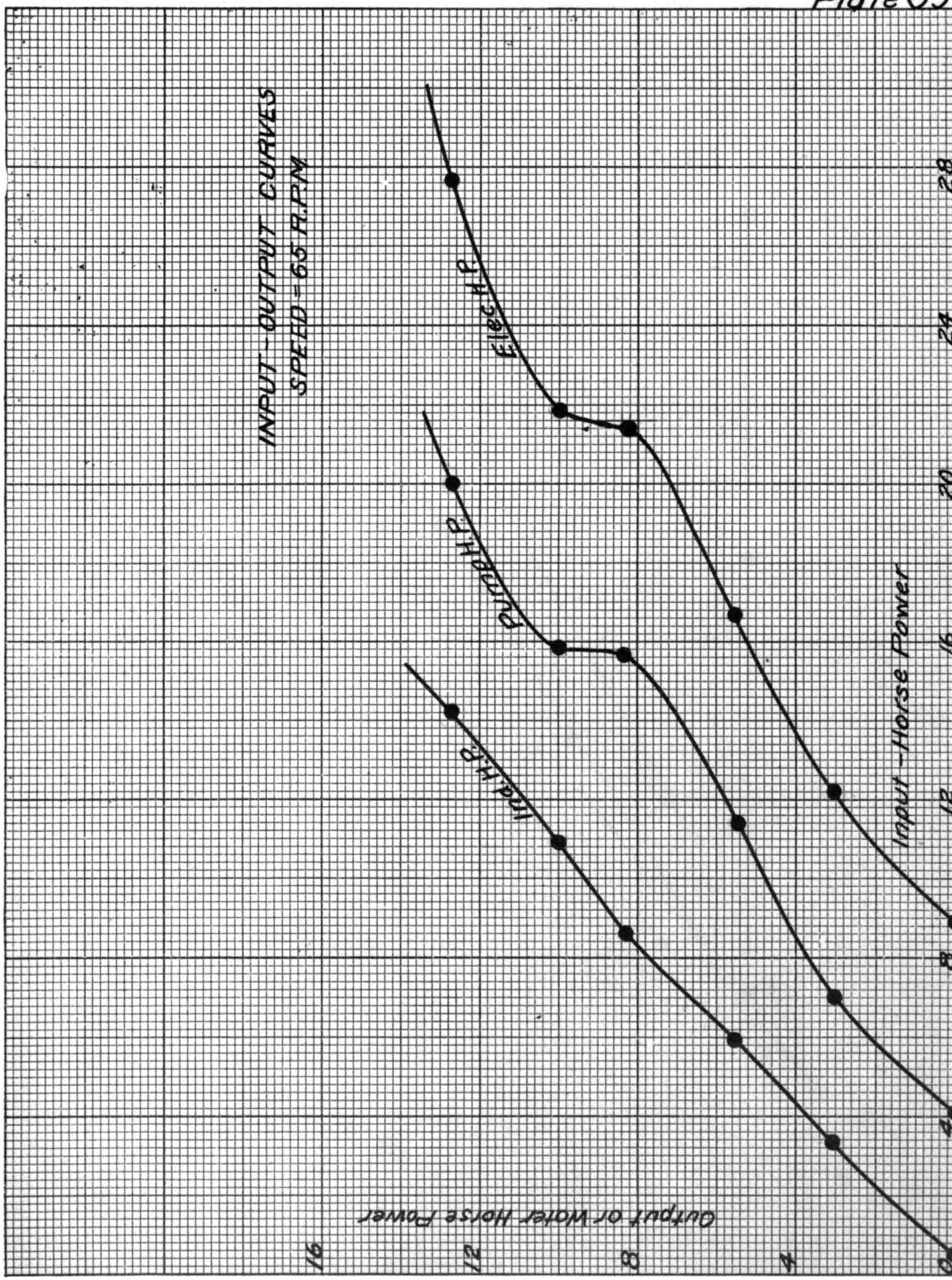
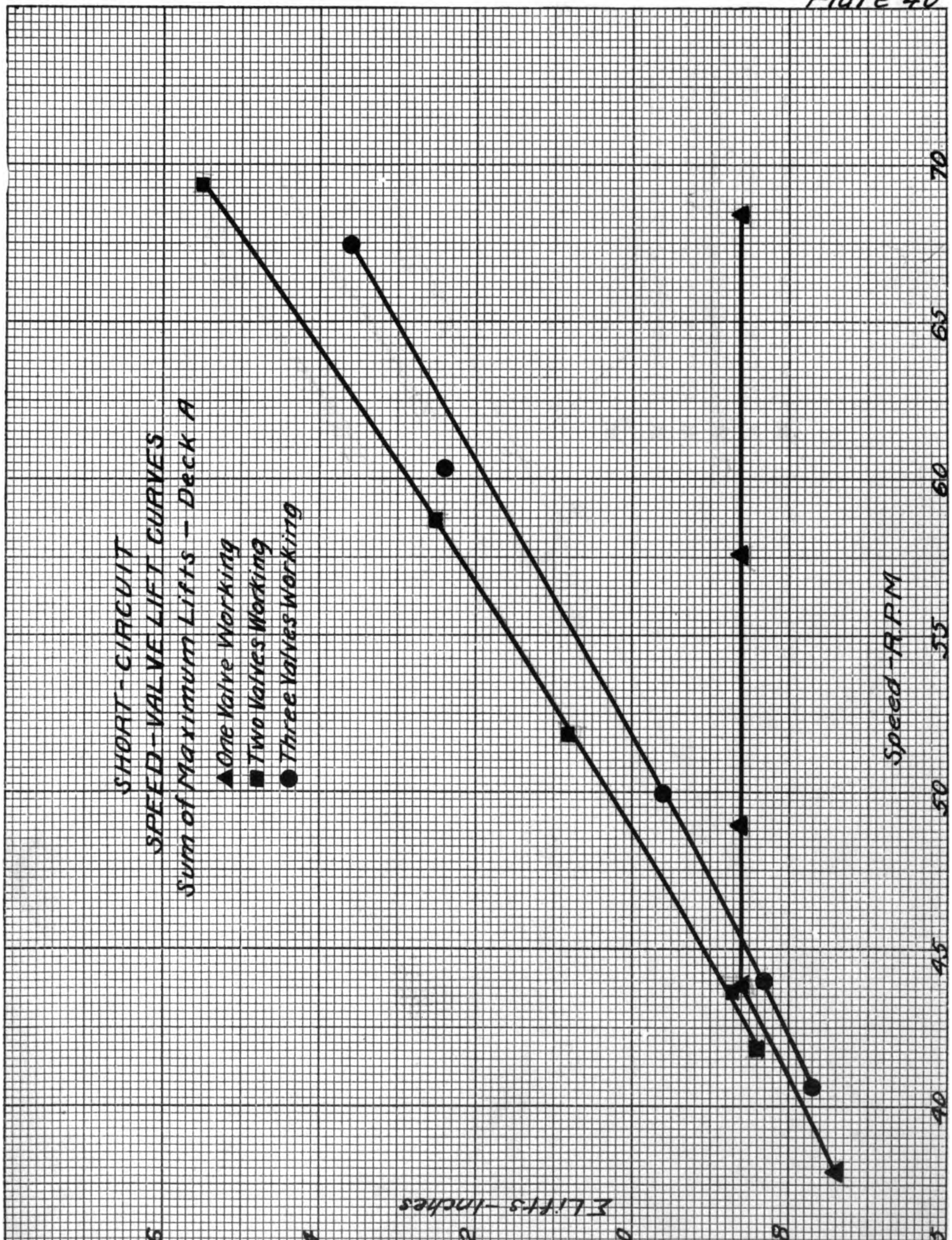

INPUT-OUTPUT CURVES
SPEED = 50 R.P.M
Elec. H.P.
Pump H.P.
Ind. H.P.
Input Horse Power
Output or Water Horse Power
0
4
8
12
16
20
4
8
12

Plate 39

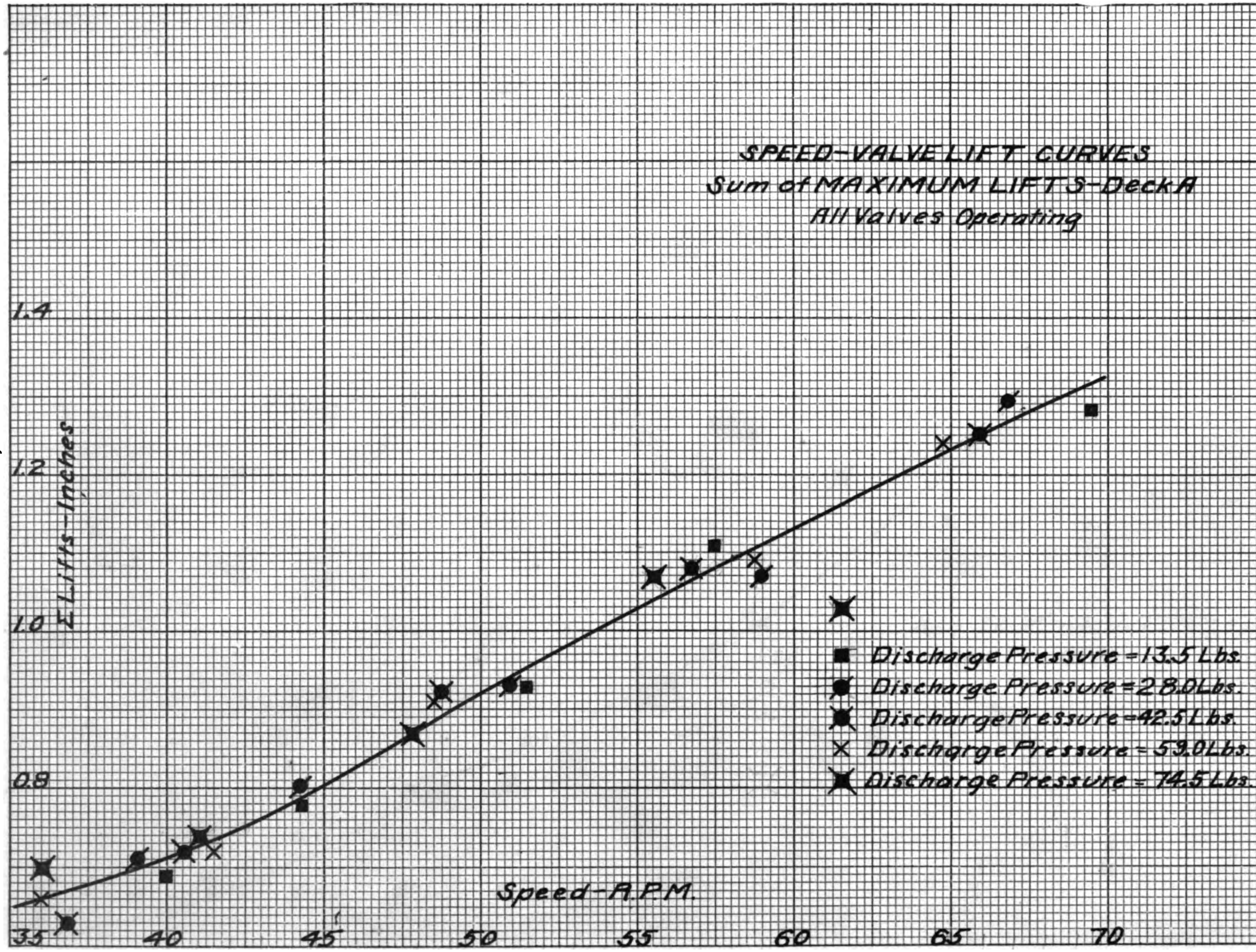


INPUT-OUTPUT CURVES
SPEED = 65 R.P.M.
Elec. H.P.
Pump H.P.
Ind. H.P.
Input - Horse Power
Output or Water Horse Power
4
8
12
16
20
24
28
0
4
8
12
16

Plate 40



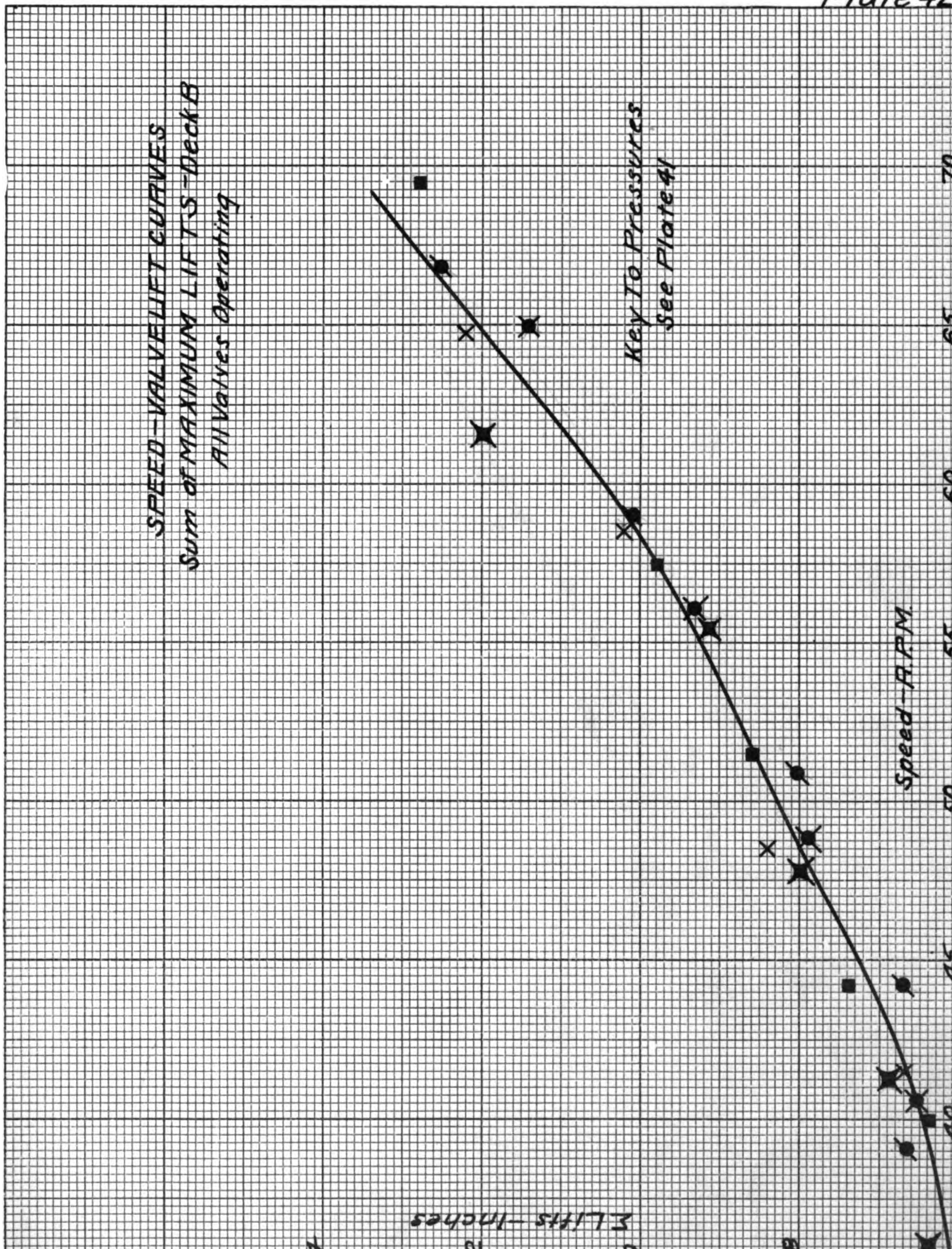
SHORT-CIRCUIT
SPEED-VALVE LIFT CURVES
Sum of Maximum Lifts - Deck A
One Valve Working
Two Valves Working
Three Valves Working
Speed-R.P.M.
Σ Lifts-Inches
35
40
45
50
55
60
65
70
0.8
1.0
1.2
1.4
1.6



SPEED-VALVE LIFT CURVES
Sum of MAXIMUM LIFTS-Deck A
All Valves Operating
1.4
1.2
1.0
0.8
Σ Lifts-Inches
35
40
45
50
55
60
65
70
Speed-R.P.M.
Discharge Pressure = 13.5 Lbs.
Discharge Pressure = 28.0 Lbs.
Discharge Pressure = 42.5 Lbs.
Discharge Pressure = 59.0 Lbs.
Discharge Pressure = 74.5 Lbs.

Plate 41

Plate 42


SPEED-VALVE LIFT CURVES
Sum of MAXIMUM LIFTS - Deck B
All Valves Operating
Key To Pressures
See Plate 41
Speed-R.P.M.
ΣLifts-Inches
40
45
50
55
60
65
70
0.8
1.0
1.2
1.4

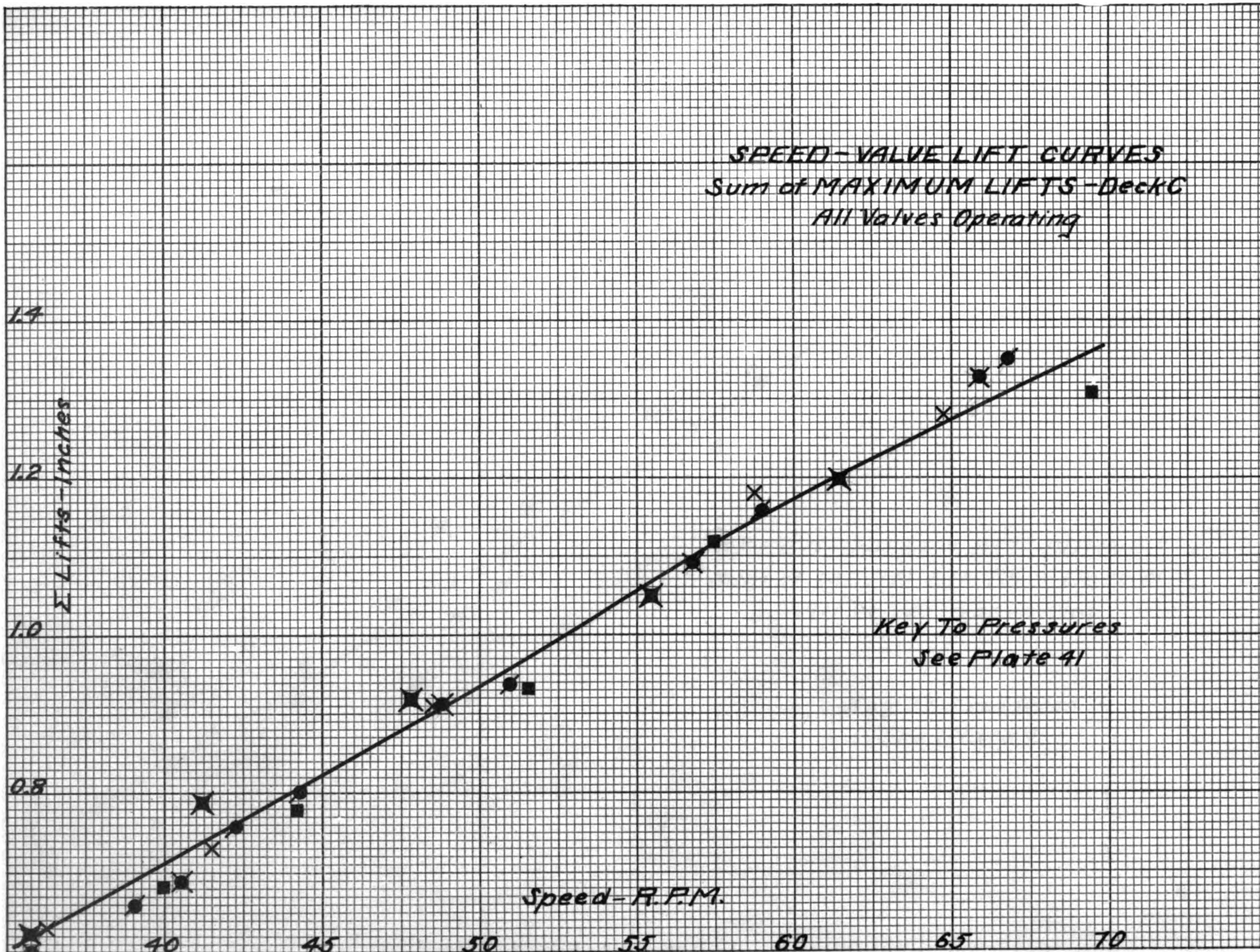
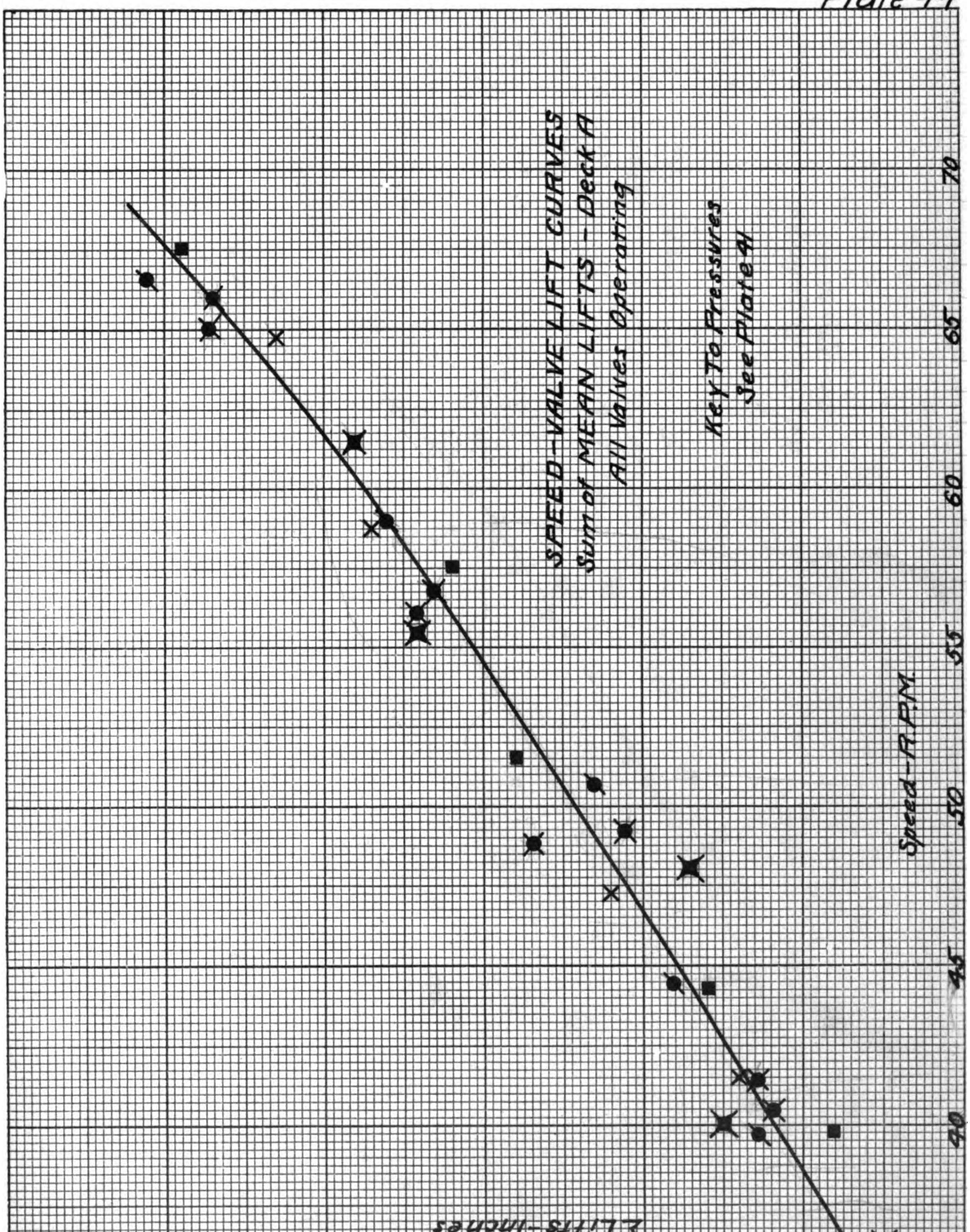
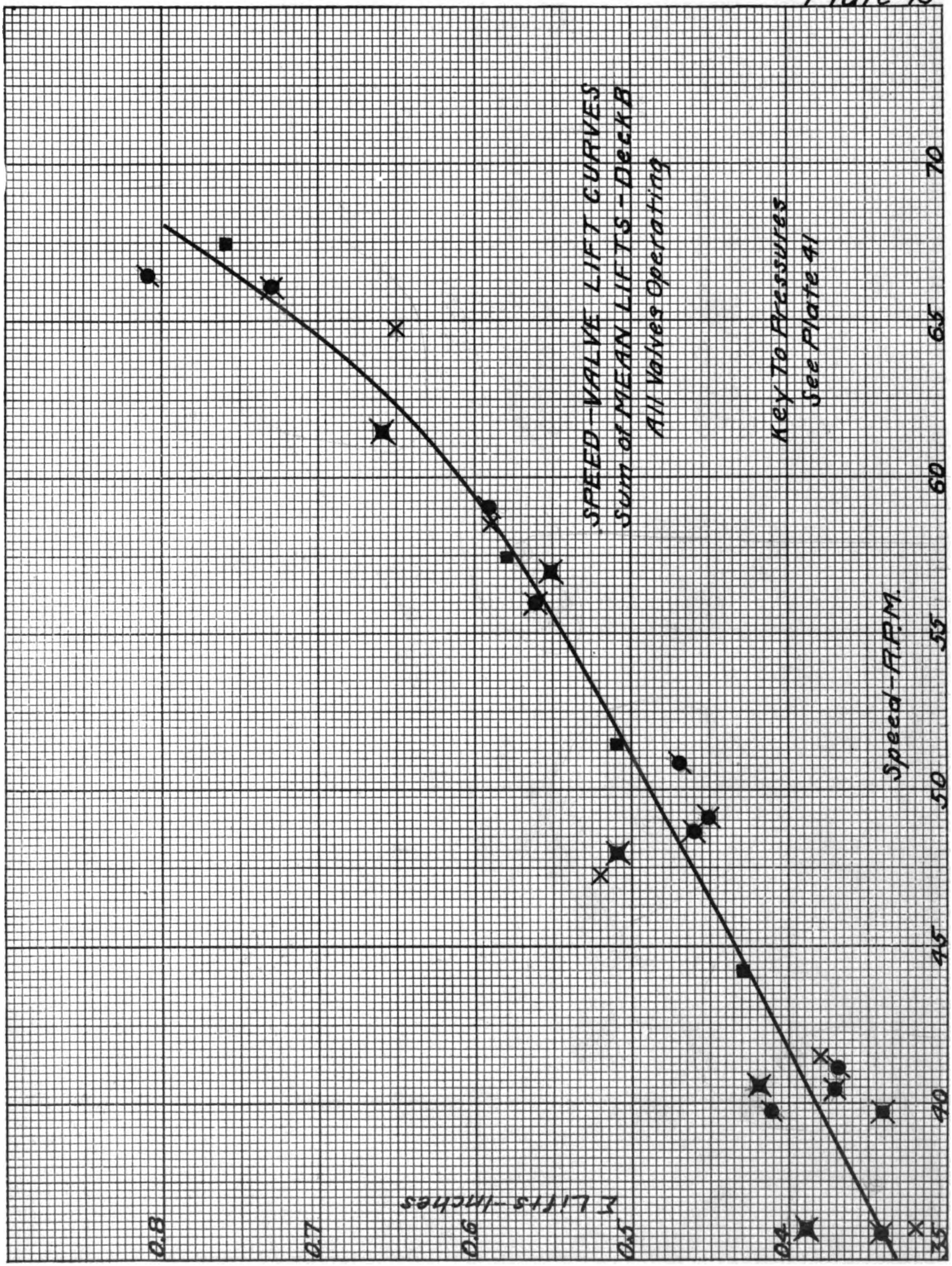

SPEED-VALVE LIFT CURVES
Sum of MAXIMUM LIFTS-Deck C
All Valves Operating
Key To Pressures
See Plate 41
Σ Lifts-Inches
1.4
1.2
1.0
0.8
Speed-R.P.M.
40
45
50
55
60
65
70
Plate 43

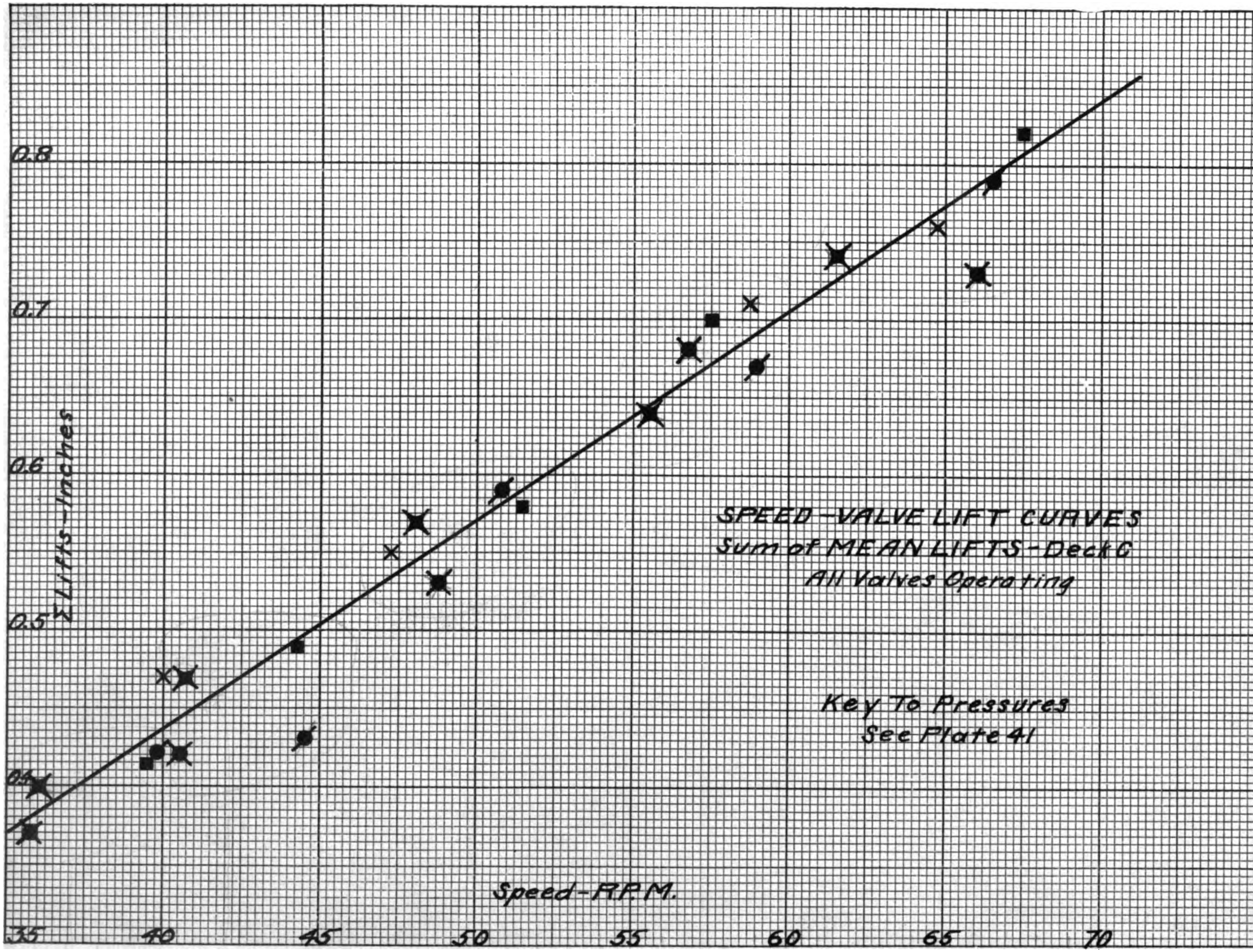
Plate 44



SPEED-VALVE LIFT CURVES
Sum of MEAN LIFTS - Deck A
All Valves Operating
Key To Pressures
See Plate 41
Speed-R.P.M.
ΣLifts-Inches
35
40
45
50
55
60
65
70
0.4
0.5
0.6
0.7
0.8

Plate 45

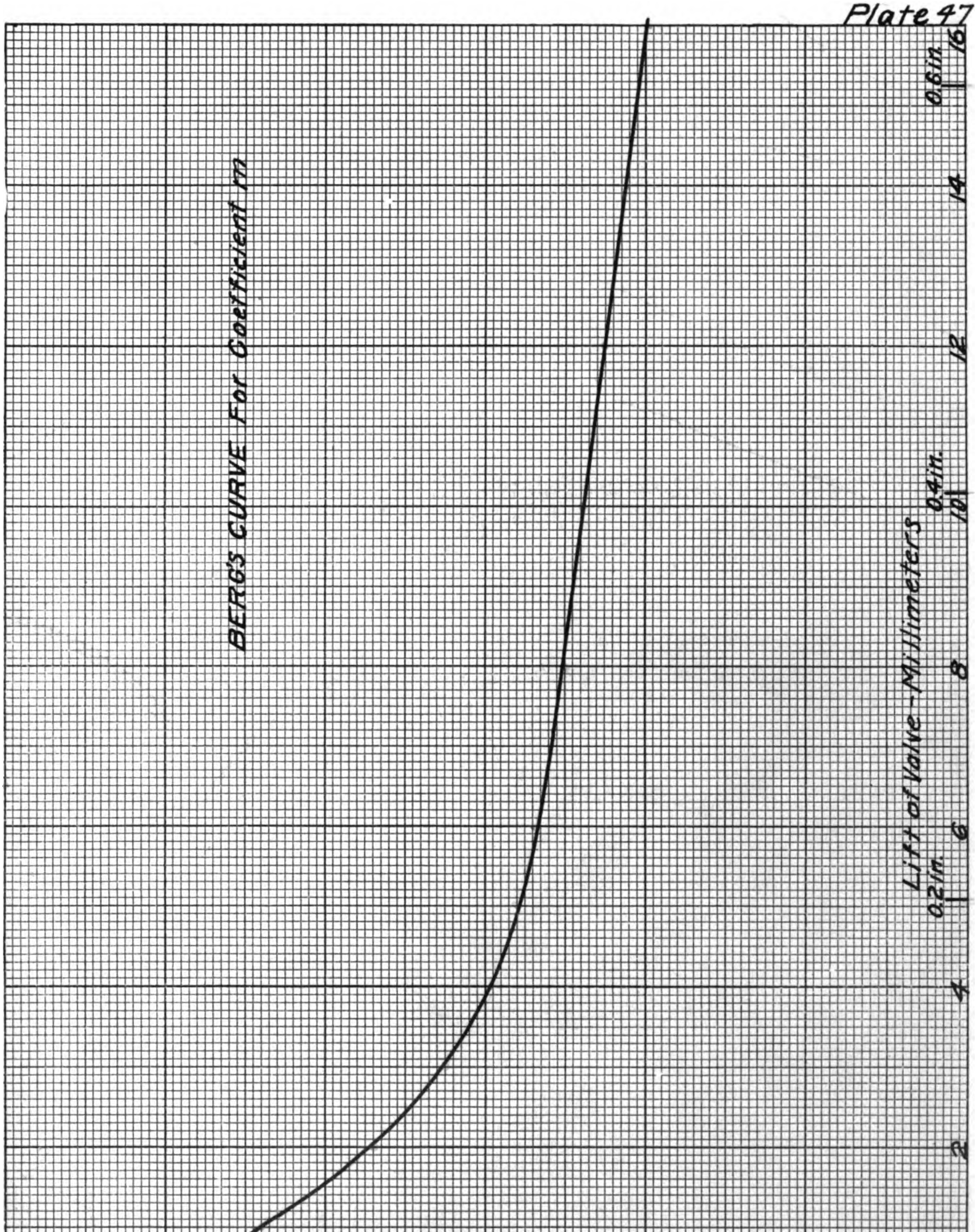

SPEED-VALVE LIFT CURVES
Sum of MEAN LIFTS - Deck B
All Valves Operating
Key To Pressures
See Plate 41
Speed-R.P.M.
Σ Lifts-Inches
0.8
0.7
0.6
0.5
0.4
35
40
45
50
55
60
65
70



SPEED-VALVE LIFT CURVES
Sum of MEAN LIFTS-Deck C
All Valves Operating
Key To Pressures
See Plate 41
Σ Lifts-Inches
Speed-R.P.M.
0.8
0.7
0.6
0.5
0.4
35
40
45
50
55
60
65
70

Plate 46

Plate 47


BERG'S CURVE For Coefficient m
Coefficient m
Lift of Valve - Millimeters
0
2
4
6
8
10
12
14
16
0.2in.
0.4in.
0.6in.
1.0
0.8
0.6
0.4
0.2

Accepted Daniel W Mead

June 1st 1911